Engineering Case Library

ECL 255

TWO SPEED COILER GEAR BOX

Peter Sawchuck is required to design a two speed gear box for a cold rolling mill coiler. He undertakes the design of a One-Off Unit that will satisfy the requirements. The basic gear design is governed by AGMA Specs. But the remaining components are governed by reliability and economic considerations.

Note: The names and places in this case have been changed but the events are based upon a real project.

TWO SPEED COILER GEAR BOX

Part A

Peter Sawchuck had just completed the design of a two-speed coiler gear box. He was pleased to see it fully assembled on the shop floor. He felt good about it and was certain that nothing could go wrong. This was important because for a unit this size there were no facilities to run the unit under design loading. Therefore, the unit had to work the first time.

Peter was in his early career. He had graduated from an eastern engineering school and was now applying for registration as a professional engineer. He had been working for Northern Gear Works for 5 years, in which time had established himself as their specialty gear designer.

Northern Gear Works was a division of a multi-national company. The parent company divided up the work on a regional basis rather than on a product line basis. They felt that any cost advantage that would be gained by quantity production would soon be offset by warehousing and shipping costs. Besides, regional responsibility meant that the divisions could respond quicker to customers' needs, and in a highly competative market customer satisfaction often made the difference between profit and loss. Consequently Northern Gear produced a mixed line of standard and special power transmission equipment.

Three months earlier Peter had received the work order (Exhibit 1) for the coiler drive. Although coiler drives were not a stock item, Peter was familiar with the application.

For steel cold rolling mills, coilers are used to reel and unreel the steel being processed. (Exhibit 2) Such coilers are capable of providing the high forward tensions that may be required in rolling, as the strip is stretch levelled and wound tightly and uniformly on the drum.

Matching the rolling and coiling speeds and maintaining constant tension in the strip are complicated by the changing diameter of the coil as it is being built up. In modern mills, the coilers are driven by variable speed motors which ensures matched speeds at the desired tension level. Moreover, in reversing mills, when the strip is being paid off a coiler, the motor acts as a generator returning energy from the back tension to the power supply.

The diameters of drums (when expanded) are usually about 16 inches. To minimize the possibility of coils collapsing or telescoping, the drum diameter should be as small as possible. However, the diameter should not be so small that the strip is plastically deformed as it is bent around it. Moreover, it must have sufficient strength to bear the weight of the strip, the effect of the strip tension and the radial pressures exerted on it by the strip. Unfortunately, the internal components of the drum required to reduce its diameter for coil removal also diminish the strength of the drum and necessitate its being of the order of 16 inches.

Mandril design for cold mills varies considerably ranging from simple drums with no mechanisms for fastening the strip to the mandril to more elaborate types that clamp the head end of the strip. With the former, a common practice is to fasten the end of the strip onto the drum with a piece of adhesive tape and then rotating the drum to accumulate several wraps on it before applying high tension.

The power required to drive a coiler is stated to be

$$\left(M_{bend} + TR + \Sigma\, P\, \mu\, \frac{d}{2}\right) \frac{\nu}{R\eta}\, \frac{1}{33{,}000}\ \text{HP}$$

Where: M_{bend} is the bending moment of the strip when uncoiling and in practical calculations is often taken as equal to the moment of plastic bending $M_S = \sigma_s S$ (pounds-feet) where σ_s is the yield stress and S is the moment of resistance of the strip to plastic bending.

T is the tension in the strip (lbs).

R is the minimum radius of the coil (practically the radius of the drum) (feet).

$\Sigma P \mu d/2$ is the total moment of friction in the bearings of the drum shaft (allowing for the weight of the drum and the coil) (pounds-feet).

ν and η are the speed of coiling (feet per minute) and the efficiency factor of the transmission.

For his registration as a professional engineer Peter was required to submit a thesis. He chose to write about the design of the coiler gear box. In it he describes the problems of designing such a gear box.

The coiler reducer had to be designed and manufactured to satisfy a number of special requirements delineated by the customer as the "Design Specifications". (Exhibit 1)

Although specifications list only the bare essentials, it is the designer's responsibility to consider many other factors whether they appear in the basic specification or not. In addition to considerations of strength, durability and efficiency he must ensure that his design meets temperature rise limitations for the anticipated range of ambient temperatures; does not exceed acceptable noise levels, meets weight limitations, and that the design is economical.

Various factors contribute to the cost of designing and manufacturing a speed reducer. These are: -

1) Quantity.
2) Size and weight of units (centre distance, diameters of gears and pinion, face width).
3) Load Rating - (surface durability and strength).
4) Type of Gearing.
5) Tooth Form.
6) Gear Ratio.

7) Shaft Configuration.
8) Gear Material and Heat Treatment.
9) Housing Design and Material.
10) Manufacturing considerations - (Gear quality specifications, tolerances and allowances, manufacturing methods, facilities available, tooling).

The initial order asked for only two coiler reducers, therefore the design must be treated as a custom built unit. However, economical considerations dictated the need to utilize as many standard components as possible, and to ensure that the reduction unit could be manufactured by established methods and within the capacity of existing machinery.

As indicated by Exhibit 3, the cost of a reducer is proportional to its weight or size. Size is controlled by pitch diameters of pinions and gears or by shaft centre distance.

It is generally assumed that the cost of a reducer will vary directly as the square of the shaft centre distance and in direct proportion to gear face width.

It is obvious from the foregoing that economy would be best served by selection of the smallest gear set capable of transmitting the required horsepower.

Helical gears were an obvious choice for the general type of gearing to be utilized in the coiler reducer. This form of gearing is most commonly used in modern applications. The main advantage of helical gears is that more teeth are in contact simultaneously and load is transferred gradually and uniformly as successive teeth come into engagement. Consequently, helical gears operate more smoothly and carry larger loads at higher speeds than do

spur gears. The helix angle usually varies between 15° and 30°.

At the conceptual stage in the design a choice had to be made between single helical and double helical arrangements. Also, it was necessary to decide whether to utilize double helical gears of the herringbone type or gap type double helical gearing. These choices involve a number of considerations.

The principal disadvantage of single helical arrangements is the generation of an axial thrust load parallel to shaft centre lines. The magnitude of the thrust force is a function of the helix angle.

In single helical applications the helix angle is kept as low as possible consistant with good tooth-contact condition in order to minimize the thrust, and to keep the thrust bearings within reasonable size. The thrust occurs at the pitch line; it produces a reaction at bearings tending to choke the axis of the pinion or gear. The helix angles of gears on a single shaft may be arranged, to create opposing thrust forces which partially cancel one another. This is one way of offsetting a portion of the disadvantage inherent in single helical gearing.

Another approach that is open to the designer is that of adjusting helix angles. Thrust force can be increased by increasing the helixal angle. Consequently, for gears on the same shaft, if the helix angle is increased for the gear train having the smaller radial force, and if the helix angle is decreased on the train having larger radial forces, this will have a compensating effect by bringing the opposing thrusts closer together. When this

approach is used, care must be taken that helix angles are kept within allowable limits, and that the alteration in thrust loads does not result in adverse conditions in adjacent shafts and gear trains.

Double helical gears are of two types. They can be manufactured with a gap between the helices or they can be of the herringbone type, where opposite hand helices are cut without a gap.

Both types of double helical gears are a combination of right and left handed helices. This results in the thrust load being fully balanced, which is a decided advantage when selecting supporting bearings. This also results in lower loads on the gear reducer housing.

One advantage of double helical arrangements is that they suffer only one half the misalignment error of a single helical gear of the same face and mounted with the same degree of error. Also, since there is no axial thrust helix angles can be maintained around 30^{O}, thereby providing a large face overlap.

The gap between helices is required when double helical gears are prepared by hobbing. The gap is necessary in order to permit run out and run in of the hob.

The gap is machined slightly below the root diameter of the gear, and gap width is governed by the diametral pitch of the gear, the outside diameter of the cutting hob and the helix angle.

An investigation of double helical gear gap type tooth arrangement was initiated for the coiler reducer. Herringbone helical gearing was not considered because the manufacturing equipment was not available for this type of gear cutting.

A preliminary check of the low speed set indicated that the gap, if double helical gears were used, would represent 23.8% of the face width. Consequently, the housings would have been wider, shaft spans would have been greater, and cost of machining would have increased. The higher machining cost was due to three hours of additional set up time required to produce the opposite hand helix.

All things considered, a cost analysis indicated that single helical gears could be produced at an overall cost saving of approximately 8%.

There was very little choice in selecting tooth form. Nearly all gears manufactured by the author's company use the standard involute tooth profile.

In order to drive a set of gears and to transmit power smoothly with minimum power loss, the mating gears must be such that, before one pair of teeth goes out of contact during mesh, the second will pick up its share of load. This is known as continuity of action. A further requirement is that the shape of the contacting surfaces of the teeth (or active profile) must be such that the angular velocity of the pair is smoothly imparted to the driven member in the proper ratio. The involute curve meets all of these requirements and therefore is most widely used as the tooth profile.

Another outstanding feature of this curve form is that the addendum and tooth thickness can be adjusted to required or corrected values. These variations of involute tooth form can be produced with standard gear tooth generating tools.

Gear ratios per set varying from unity to 10 : 1 are customary in ordinary spur or helical gear reducers. For each gear set a shaft, two bearings, shims for adjustment and bearing retainers are required. When large ratios are needed, the choice of gear arrangement is often determined by finding the combination with the fewest practical number of parts that will do the job adequately.

Practical considerations limit single reduction units to a ratio of 10:1, double reduction units to 40:1 and triple reduction helical gear reducers to ratios of about 300:1.

In selecting reducers for steel mills, reliability is usually the prime consideration. Failure of a reducer can result in losses amounting to thousands of dollars per day. Parallel shaft configuration was chosen for this reducer because of its ability to withstand heavy loads, temporary overloads and heavy continuous service.

The load rating of helical gears is generally determined by surface durability and strength requirements. The relative importance of these characteristics is shown in Exhibit 4.

Surface durability can be defined as the load that can be carried without damaging the profile of the tooth.

There are many types of surface destruction, for example, corrosive wear, abrasive wear, scoring, pitting and spalling. Pitting is by far the most frequent form of surface destruction and is considered a fatigue type of failure. If the load is high enough the surface of the tooth will be eaten away with pits after some millions of cycles. This rolling and sliding contact fatigue phenomenon is not fully understood. One theory is that the formation of pits is due to the lubricant entering small surface cracks and developing a hydraulic action that lifts fragments of metal from the surface.

Strength rating can be defined as the load that the tooth can carry without permanent deformation or fracture. The fractures can be due to heat treatment and grinding, stress concentrations and bending fatigue.

Exact computation of load carrying capacity is a rather difficult process because of the number of variables involved. Typical of these are gear cutting accuracy, mounting errors, elastic deflection, material, quality, tooth stiffness, and characteristics of driving or driven machinery.

There are standard procedures for computing gear ratings, best known of which are those of Lewis, Buckingham, Barth, Dolan-Broghamer, Gleywood, Kelley-Pedersen and AGMA. Surface durability and strength rating for gears are calculated according to the AGMA standard at Northern Gear Works.

Considerable attention must be given to the profile and longitudinal correction in designing the teeth for large gears such as the coiler reducer gears.

In order to avoid shocks, as gear teeth enter or leave the field of engagement the flank profile is eased back locally. This is known as "tip and root relief". (Exhibit 5 & 6) This modification compensates also for elastic deflection of the loaded teeth and avoids involute interference. An investigation conducted by H. Sigg indicated that the tooth interference on the loaded true involute gear is:

$$\delta s = \text{(approx.) } 5 \times 10^{-3}\ Wg$$

where Wg = normal gear force along the line of action in lbs/in.

δs = displacement in ten thousandths of an inch

(See Exhibit 6)

The form and degree of profile correction is controlled accurately on grinding or hobbing machines, and the relieved area is blended smoothly into the remaining true involute area.

Nearly all hobs for generating involute gears have built in capacity for tip and root relief.

H. Sigg's formula for estimating degree of correction for Helical Gears at first point of tooth contact is:

$$\Delta_u = 2 + 2.8\ W \times 10^{-3} \quad \text{for lower tolerance limit}$$

$$\Delta_o = 5 + 2.8\ W \times 10^{-3} \quad \text{for upper tolerance limit.}$$

where

Δ = correction in ten thousands of an inch.

W = peripherical unit load in lbs. per one inch of face width.

Standard hobs will relieve .001 of the tip and root of the gear teeth; further profile modification may not be required.

Even the most accurate and correctly machined gear teeth are subject to gear tooth inaccuracies, shaft deflection, housing machining error, manufacturing misalignment, hardening distortions, all of which impair the running behavior of the gears and cause undesirable noise and excessive wear. In order to alleviate or minimize the effects of these variations longitudinal correction or crowning was applied to high, first intermediate, and low speed pinions only. (Exhibit 7)

Crowning leaves the tooth thinner at the ends and provides for central tooth loading over about 85% of the face width. The amount of crowning for standard commercial gears is about .003" per side of the tooth.

The longitudinal correction for the coiler reducer has to be determined by calculating pinion elastic deflection. The resultant gear tooth form is shown in Figure 8.

With these considerations in mind, Peter was able to design the two speed coiler. Once the essential components were selected and the layout drawn, he had to give serious consideration to secondary but still vitally important factors as:

Housing Design - *Cast or fabricated*

Thermal Rating - *Horse power transmitted continuously without overheating (70° F temp. rise to maximum of 200° F)*

Lubrication - *Oil selection and distribution*

Seals

Noise

Assembly and handling

Exhibit 1-a

WORK ORDER

Date: January 9th, 1972

Work Order No.: 936852

Customer: Colossal Consultants Inc.

Quantity: 2

Description: 100 H.P. Two Speed Coiler Drive Gear Box

DESIGN SPECIFICATIONS

Prime Mover:	100 H.P. DC Motor
Design Speed:	850 r.p.m.
Maximum Speed:	2000 r.p.m.

COILER REDUCER

Dual output speed gear reducer	38.7 & 12.9 output r.p.m.
Gear ratios:	66 to 1 and 22 to 1
Mechanical rating:	100 H.P. at AGMA S.F. 1.0
Thermal rating:	100 H.P.
Gear shifter:	Manual
Length of low speed shaft:	58"
Shaft to support:	52" wide x 72" dia. coil maximum 30,000 lbs.

SPECIAL INSTRUCTIONS:

These units are part of a cold rolling mill being supplied to Brazilian Enterprises. It is expected that at a later date additional units will be required. Gear set is to be designed to AGMA specifications. Preliminary sketch of gear box is attached based on a previous design. Foundation mounting will be in accordance with these dimensions.

Exhibit 1-b

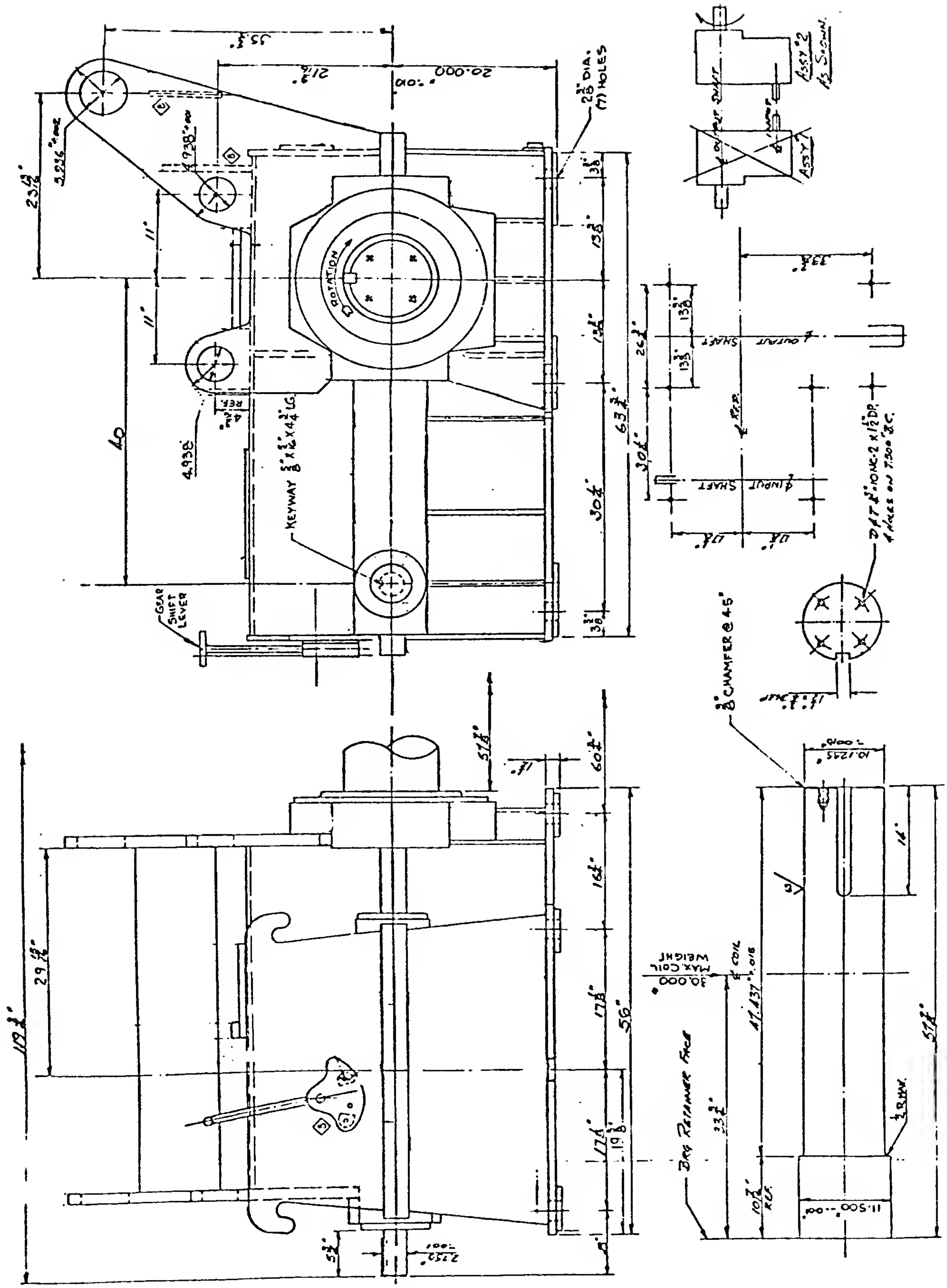
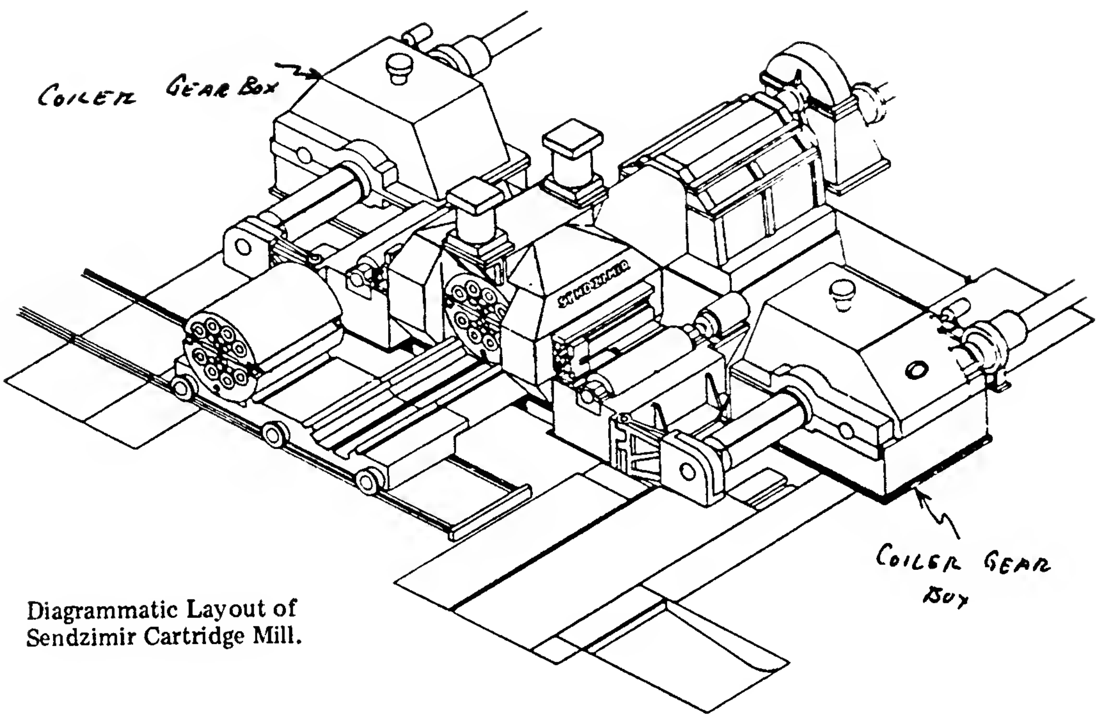
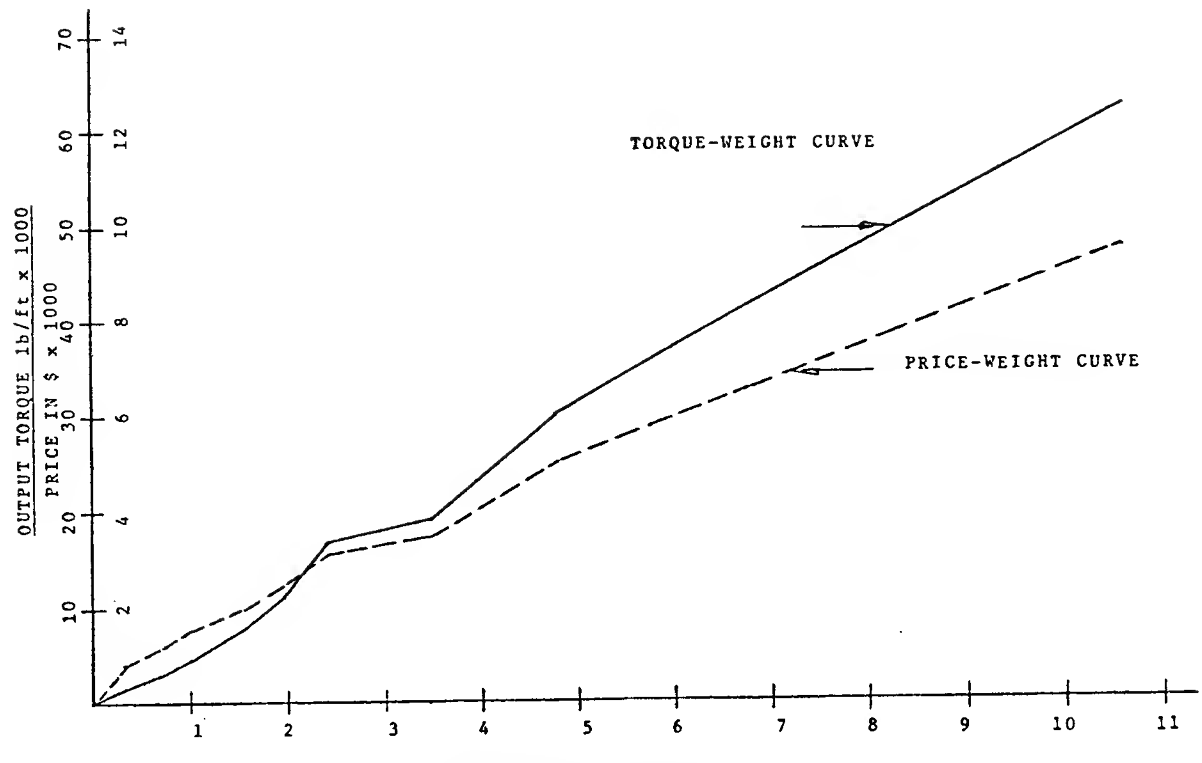

GEAR SHIFT LEVER
ROTATION
INPUT SHAFT
OUTPUT SHAFT
MAX. COIL WEIGHT
BRG RETAINER FACE
ASSY 1
ASSY 2 AS SHOWN

Exhibit 2

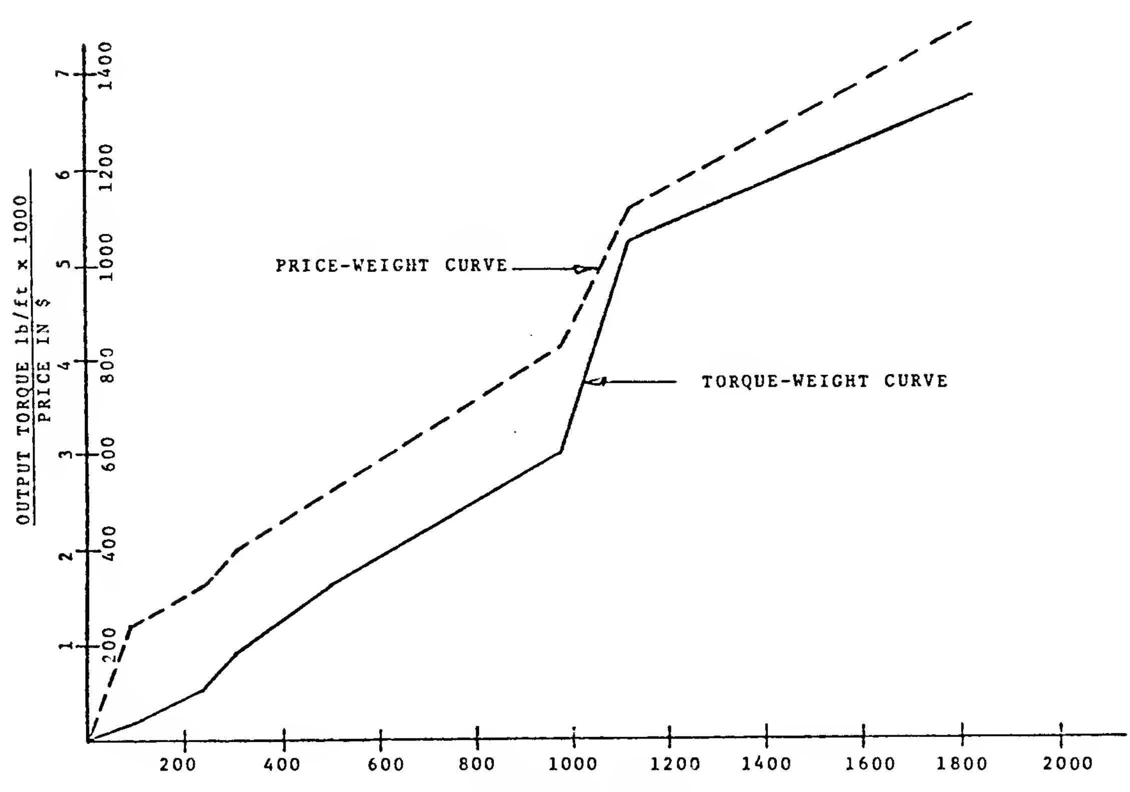


Diagrammatic Layout of
Sendzimir Cartridge Mill.

Exhibit 3

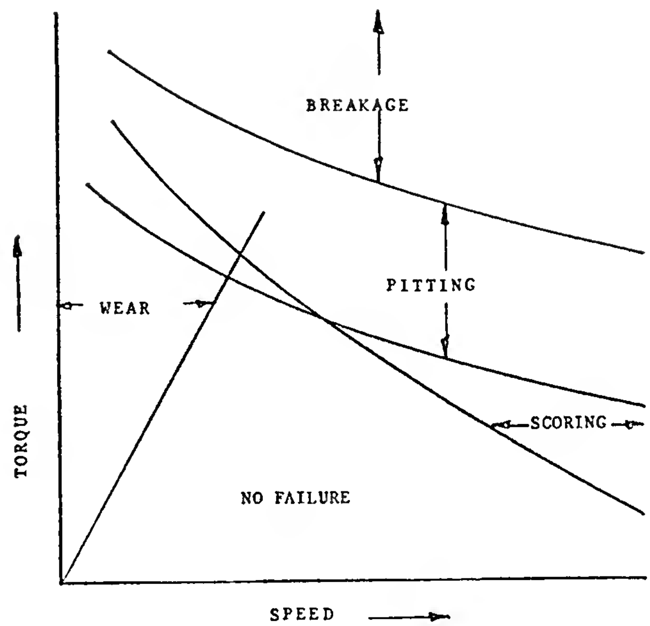


DOUBLE RATIO PARALLEL SHAFT SPEED REDUCERS



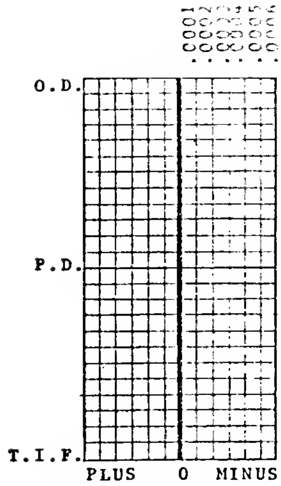
IN-LINE HELICAL GEAR SPEED REDUCERS

Exhibit 4

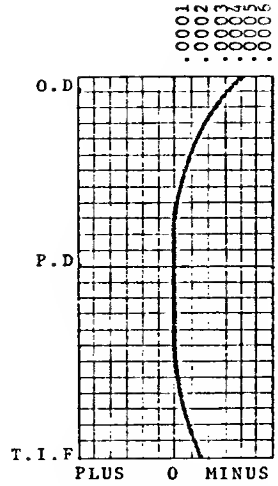


GEAR FAILURE CHARACTERISTICS

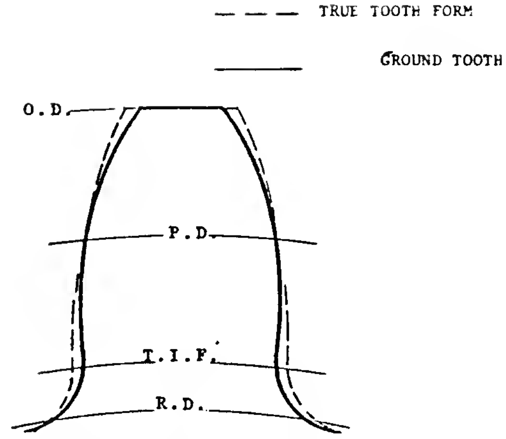
SOURCE: "LUBRICATING GEARS"
BY: R.R. Lorvick
Machine Design, July 1970



INVOLUTE CHART SHOWING TRUE INVOLUTE FORM



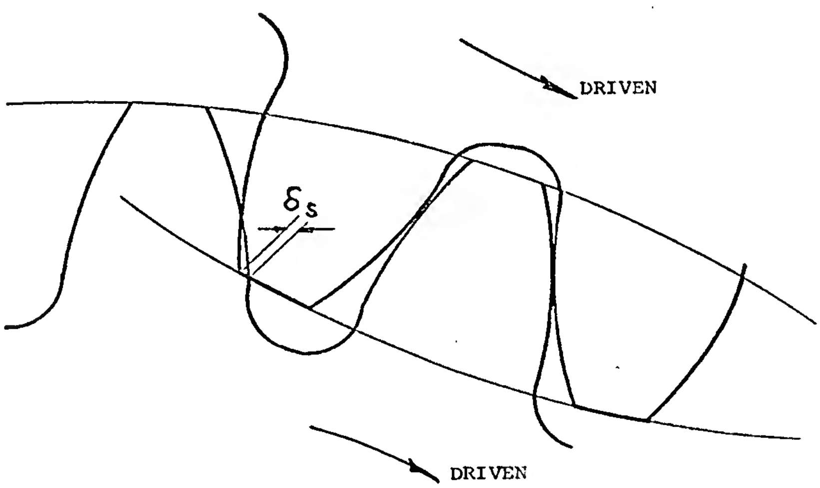
INVOLUTE CHART SHOWING DEVIATION FROM TRUE INVOLUTE



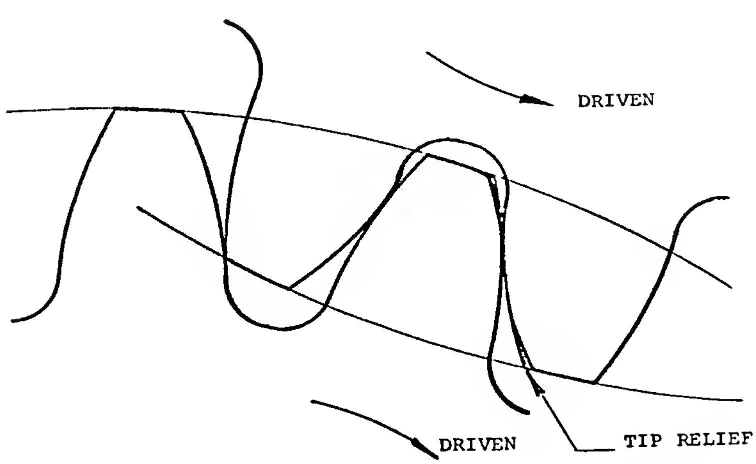
SECTION "BB" SHOWING MODIFICATION IN INVOLUTE FORM (EXAGGERATED).

Exhibit 5

INVOLUTE SECTION "BB"

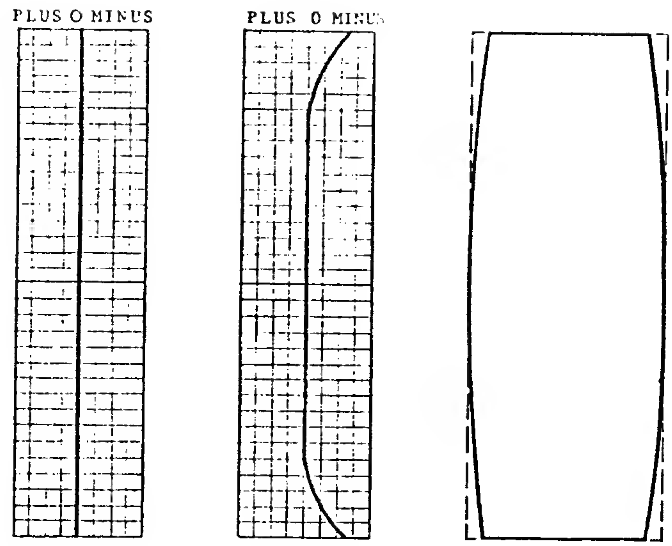


TOOTH INTERFERENCE ON LOADED TRUE-INVOLUTE GEAR



LOADED TRUE INVOLUTE GEAR WITH PROFILE CORRECTED

Exhibit 6



LEAD CHARD
SHOWING UNCROWNED
LEAD FORM

LEAD CHART
SHOWING CROWNED
LEAD FORM

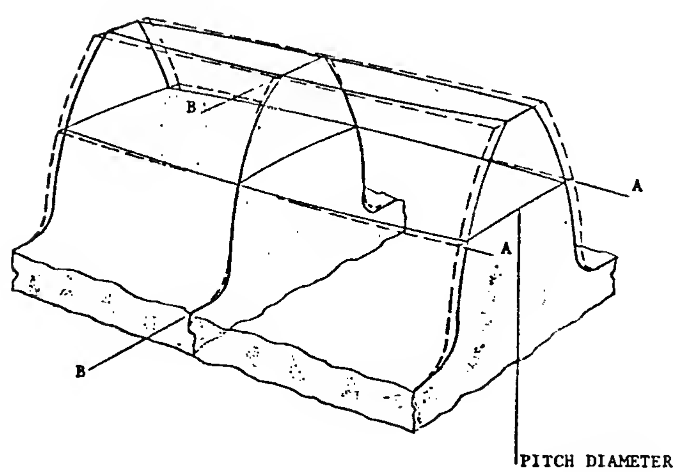
SECTION "AA"
SHOWING LEAD
CROWN (EXAGGERATED)

LEAD SECTION "AA"

Exhibit 7

— — — TRUE TOOTH FORM

——— CROWN GROUND TOOTH
FOR (EXAGGERATED)



TOOTH FORM FOR CROWN-GROUND HELICAL GEARS

Exhibit 8

TWO SPEED COILER DRIVE

Part B

Peter Sawchuck started his design by making some simple sketches of the gear and shaft arrangements (Exhibit 9). Because the maximum reduction ratio was to be 66:1 he settled on a triple reduction gear train, which meant 4 shafts with bearings, etc. To provide lower reduction of 22:1 he decided to use a simple jaw clutch on shaft No. 2. By shifting the clutch axially the clutch engaged either the low speed or high speed first reduction. This of course required that gears on shaft 2 be mounted on bearings so that they could rotate independently when not engaged by the clutch. To provide the smallest pinions possible and to have very rigid shafts, at each stage the pinion is machined integral with the shaft.

Once the basic layout was settled all that remained was to work out the details. In his report Peter illustrates the detail analysis and decisions necessary to arrive at a final design.

The power rating of the gear set is primarily based on the surface durability of gear teeth. Table 1 shows typical relation between the strength and surface durability for helical gears.

Compressive strength is the key factor influencing surface durability. The compressive stress at tooth contact is proportional to the square root of the load. For 20° teeth this relation based on Hertz stress equation is:

$$Sc = 5715 \sqrt{\frac{Wt}{Fd} \left(\frac{m_g + 1}{m_g}\right)}$$

Where Sc = compressive stress, p.s.i.

Wt = tangential force

F = net contact face width

m_g = speed ratio

d = pinion pitch diameter

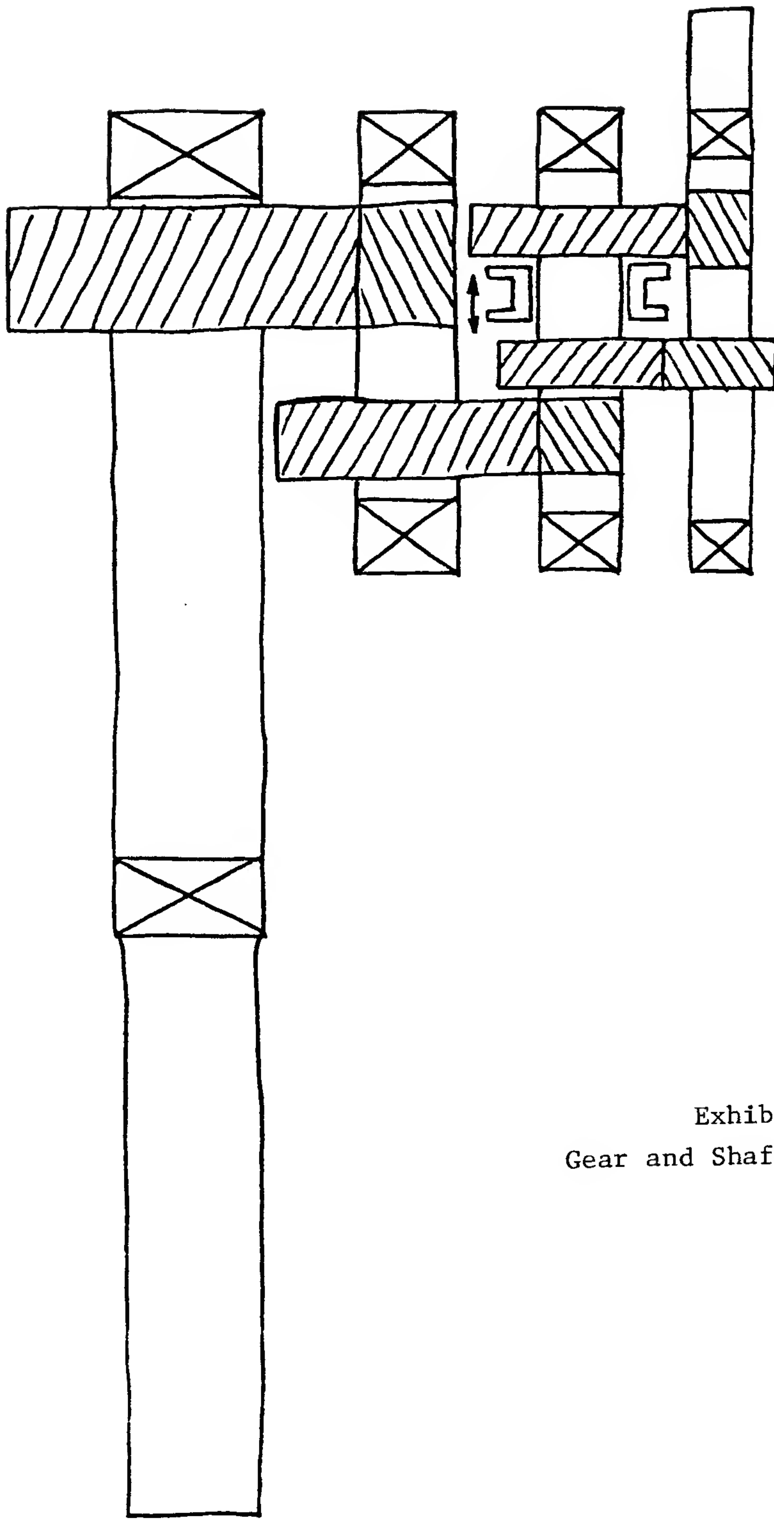


Exhibit 9

Gear and Shaft Arrangement

TABLE NO. 1

TYPICAL SURFACE STRESS CAPACITY OF HELICAL GEARS

PINION STEEL		GEAR STEEL		SURFACE STRESS CAPACITY
Carbon	220 Bhn	Carbon	180 Bhn	1.0
Alloy	300 Bhn	Alloy	250 Bhn	1.5
Nitrided	55 Rc	Nitrided	250 Bhn	1.8
CH & G *	60 Rc	CH & G	250 Bhn	1.8
Nitrided	55 Rc	Nitrided	55 Rc	3.0
CH & G	60 Rc	CH & G	60 Rc	3.5

* Carburized Hardened and Ground

TABLE NO. 2

STRENGTH/SURFACE CAPACITY RATIO FOR COMMON GEAR STEEL

STEEL	PINION ULTIMATE TENSILE (psi)	GEAR STEEL	ROOT STRENGTH CAPACITY	STRENGTH CAPACITY / SURFACE CAPACITY
Carbon	107,000	Carbon	1.0	5.0
Alloy	148,000	Alloy	1.25	4.2
Nitrided	148,000 core	Alloy	1.25	3.5
CH & G	148,000 core	Alloy	1.5	4.2
Nitrided	148,000 core	Nitrided	1.25	2.1
CH & G	148,000 core	CH & G	1.9	2.7

Source: "Large Power Transmission Gears"
By: C.E. Gay
Machine Design, Apr. 1972.

From the above basic stress relation Dudley developed a simple formula for establishing the smallest gear set for given horsepower:

$$Sc = C_k \sqrt{K \quad C_d}$$

$$\text{Where K factor} = \frac{Wt}{Fd} \left(\frac{m_g + 1}{m_g}\right)$$

C_k = contact stress factor which takes into consideration the point of contact between mating gears and elasticity of the material.

C_d = overall derating factor for surface durability which takes into consideration finish of surface, size, load distribution and dynamics.

From tabulated values for C_k and C_d and known material compressive stress Sc the value "K" can be computed. From the "K" factor equation one can determine gear size factor "Fd". This product is then used to determine values of face width and pitch diameter for the pinion. Using Dudley's approach, the pinion size for the coiler reducer was determined with the following considerations.

A high number of teeth means a small tooth size and thus danger of tooth breakage. For a gear hardness of 300 BHN, D.W. Dudley recommends a minimum of 22 to 23 teeth in the pinion. It is our experience that a somewhat smaller number of teeth can be used safely. Consequently a 20 tooth pinion was used.

The acceptable range of F/d ratios vary between F = d/4 and F = 2d. From this range it has been our experience that F/d ratios between F = d and F = 1.5d are most compatible with economics and available manufacturing facilities.

The most serious problem involved with gearing having small numbers of teeth is that of undercutting. An undercut tooth is one in which a portion of the profile in the active zone has been removed by cutting action.

In order to avoid this condition the gear can be produced with modified addendum. Usually the pinion is made with long addendum and the gear with short addendum. Latest studies indicate that modification of addendum helps to achieve a better condition of surface contact, improves relative motion and better interaction between the gear and the pinion.

It is common practice for gear ratios up to 3 to 1 to use an addendum ratio of 1 to 1. For higher gear ratios an addendum ratio of 3 to 1 is usually selected.

A properly designed helical gear set should have an overlap greater than 1.25. From the foregoing it is apparent that face width selected is adequate.

For gears in mesh, contact between two teeth begins at the point where the addendum circle of one gear crosses the pressure line and ends where the addendum circle of the other gear intersects this line.

In order to obtain smooth and even transfer of load from one pair of teeth to the next, it is recommended that contact ratio "m" be not less than 1.4.

After finding pinion and gear proportions using Dudley's method the surface durability and tooth strength were checked according to A.G.M.A. 211.02 and 225.01.

The following are reconstructions of the calculations (abstracted from the thesis) made for first stage gear set. The remaining gear sets were determined in the same manner. The results are given in Tables 3 and 4.

20 JAN 72

FIRST STAGE 66/1 RATIO

RECONSTRUCTED CALCULATIONS

SMALLEST GEARSET. REF. DUDLEY, PROD. ENG. AUG. 1968

$$S_c = 5715\sqrt{\frac{W_t}{Fd}\left(\frac{m_g+1}{m_g}\right)}$$

$$K = \frac{W_t}{Fd}\left(\frac{m_g+1}{m_g}\right)$$

THEN $S_c = C_k\sqrt{K C_D}$

FOR FIRST STAGE HI REDUCTION

TORQUE $T = \frac{63025 \text{ H.P.}}{N}$

$$T = \frac{63025 \times 100}{850} = 7400 \text{ in-lbs}$$

$$W_t = \frac{2T}{d}$$

$$= \frac{(2)(7400)}{d}$$

$$= \frac{14800}{d}$$

K - FACTOR ASSUME GEAR RATIO 4.4/1

$$K = \frac{W_t}{Fd}\,\frac{(m_g+1)}{m_g}$$

$$= \frac{14800}{Fd^2}\left(\frac{4.4+1}{4.4}\right)$$

$$= \frac{18180}{Fd^2}$$

20 JAN 72

FIRST STAGE 66/1 RATIO

RECONSTRUCTED CALCULATIONS

CHOOSE

PINION MAT'L SAE 43L40 HARD. 300 BN

$S_c = 130{,}000$ psi   DUDLEY TABLE VIII

$N_P = 20$

$N_G = 20 \times 4.4 = 88$

$C_K = 4365$ DUDLEY TABLE II

$C_D = 2.8$ DUDLEY TABLE IV

NOW $S_c = C_k \sqrt{K C_D}$

$$K = \frac{S_c^2}{C_k^2 C_D}$$

$$= \frac{130{,}000^2}{4365^2 \times 2.8}$$

$$= 318$$

BUT $K = \dfrac{18180}{Fd^2}$

$$\therefore Fd^2 = \frac{18180}{318}$$

$$= \underline{57}$$

CHOOSE $d < \dfrac{F}{d} < 1.5d$

$$1.25\, d^3 = 57$$

$$d = \left(\frac{57}{1.25}\right)^{1/3}$$

$$d = 3.5 \text{ in.}$$

20 MAR 72

FIRST STAGE 66/1 RATIO

RECONSTRUCTED CALCULATIONS

$F = 1.25 \times 3.5$

$= \underline{\underline{4.5 \text{ in}}}$ *

GEAR $d_g = d_p \times m_g$

$d_g = 3.5 \times 4.4$

$= 15.4 \text{ in}$

CENTER DISTANCE $C = \dfrac{d_p + d_g}{2}$

$C = \dfrac{3.5 + 15.4}{2}$

$= 9.45$ USE $\underline{\underline{9.5 \text{ in.}}}$ *

ASSUME HELIX ANGLE $\psi = 18°$

NORMAL DIMETRAL PITCH $P_n = \dfrac{N_g + N_G}{2C \cos\psi}$

$P_n = \dfrac{20 + 88}{(2)(9.5)\underbrace{\cos 18°}_{.951}}$

$= 5.95$ USE $\underline{\underline{P_n = 6}}$ (FOR STANDARD CUTTER) *

$\therefore \cos\psi = \dfrac{20 + 88}{(2)(9.5)(6)}$

$\cos\psi = 0.94736$

$\psi = \underline{\underline{18° - 40' \; 18''}}$ *

log 108 = 2.0334238
log 12 = 1.0791812
10 + 10.9542426 -10
log 9.5 = 1.9777236
9.9765190 -10

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FIRST STAGE 66/1 RATIO

$$d_p = \frac{2 \times 9.5 \times 20}{108}$$

$= \cancel{3.15852 \text{ in.}}$ 3.51852 *

$$d_g = \frac{2 \times 9.5 \times 88}{108}$$

$= 15.48148$ in. *

CHECK $\cancel{3.15852}$ 3.15852 + 15.4818 = 19.000 O.K.

USING ADDENDUM RATIO OF 3/1

PINION ADDENDUM $= (3/4)(2)\left(\frac{1}{P_n}\right)$

$= (0.75)(2)(1/6) = 0.25$ in *

GEAR ADDENDUM $= (0.25)(2)(1/6) = 0.0833$ in. *

PINION O.D. = 3.518 + (2×0.25) = 4.018 in
GEAR O.D. = 15.481 + (2 × 0.0833) = 15.648 in.

MINIMUM FACE WIDTH FOR ONE OVERLAP F'

$$F' = \frac{\pi}{P_n \sin \phi}$$

$$\text{OVERLAP} = \frac{F}{F'}$$

$$F' = \frac{3.1416}{6 \sin 18^\circ 40' 18''}$$

$= 1.64''$

$$\text{OVERLAP} = \frac{4.5}{1.64} = 2.75 \quad \text{O.K.}$$

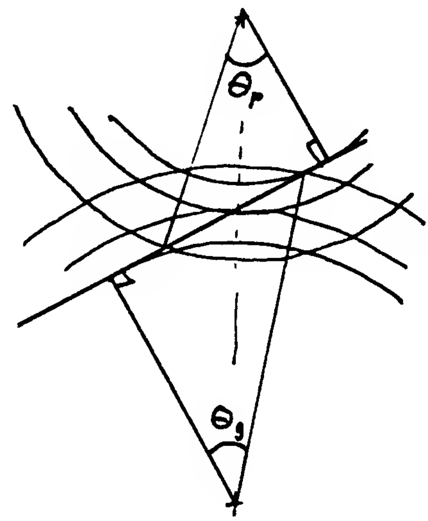
RECONSTRUCTED CALCULATIONS

FIRST STAGE 66/1 RATIO

DETERMINE IF CONTACT RATIO $m > 1.4$

$$m = \frac{u_p + u_g}{P_b}$$

ϕ : PRESSURE ANGLE
$\phi = 20^\circ$



$$u_p = r_{ag} \sin \theta_g - r_g \sin \phi$$
$$u_g = r_{ap} \sin \theta_p - r_p \sin \phi$$

BASE PITCH $P_b = \frac{\pi}{P} \cos \phi$

OUTSIDE RADIUS $r_{ap} = \frac{4.018}{2} = 2.009$

$$r_{ag} = \frac{15.648}{2} = 7.824$$

RADIUS OF BASE CIRCLES $r_{bp} = r_p \cos \phi$

$$r_{bp} = 1.759 \cos 20^\circ = 1.65235$$
$$r_{bg} = 7.740 \cos 20^\circ = 7.27322$$

$$\cos \theta_p = \frac{r_{bp}}{r_{ag}}$$

$$\theta_p = \cos^{-1} \frac{1.65235}{2.009} = 34^\circ 40' 0''$$

$$\theta_g = \cos^{-1} \frac{7.27322}{7.824} = 21^\circ 36' 30''$$

$$u_p = (7.824) \sin 21^\circ 36' 30'' - \frac{15.48148}{2} \sin 20^\circ$$

$$= 0.2338$$

$$u_g = (2.009) \sin 34^\circ 40' 0'' - \frac{3.518}{2} \sin 20^\circ$$

$$= 0.541108$$

$$u_p + u_g = 0.2338 + 0.54118$$

$$= 0.7749$$

RECONSTRUCTED CALCULATIONS

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6 of 8 FIRST STAGE 66/1 RATIO

BASE PITCH (CIRCULAR) = $\frac{d_p \pi}{N_p} \cos\phi$

$$P_b = \frac{(3.518)\pi}{20} \cos 20^\circ$$

$$= 0.5193$$

$$m = \frac{0.7749}{0.5193} = \underline{\underline{1.492}} \quad O.K. \quad > 1.4$$

CHECK SURFACE DURABILITY PER A.G.M.A. 211.02

$$S_c = C_p \sqrt{\frac{W_t C_o}{C_v} \frac{C_s}{d F} \frac{C_m C_f}{I}} \qquad C_s = C_f = C_o = C_t = C_r = 1.0$$

AND $$S_c \leq S_{ac} \left(\frac{C_L C_H}{C_T C_R} \right)$$

$$C_p = \sqrt{\frac{k}{\pi \left(\frac{1-\mu_p^2}{E_p} + \frac{1-\mu_g^2}{E_g} \right)}}$$

USE SIMPLIFIED AGMA 211.02A FORMULA

POWER $P_{ac} = C_1 \; C_2 \; C_3 \; C_4$ VALUES FROM CHARTS

$$C_1 = \frac{n_p d^2 C_v}{126{,}000} = .06$$

$$C_2 = \frac{F}{C_m} = 3.3$$

$$C_3 = 0.255 \left(\frac{m_g}{m_g+1} \right) \left(\frac{S_{ac}}{C_p} \right)^2 = ?$$

$$C_4 = C_L = 1.0$$

$$P_o = 100 \text{ H.P.}$$

RECONSTRUCTED CALCULATIONS

FIRST STAGE 66/1 RATIO

TO FIND $C_3 = \dfrac{100}{(.06)(3.3)(1.0)} = 505$

$m_g = 4.4$

MINIMUM MAT'L HARDNESS FROM TABULATED DATA

GEAR $H_B = 270$ PINION $H_B = 310$

CHOOSE GEAR SAE 4140 THRU HARDENED 270-300 H_B

PINION SAE 43L40 THRU HARDENED 310-340 H_B

CHECK TOOTH STRENGTH A.G.M.A. 225.01

$$S_t = \frac{W_t K_o}{K_v} \frac{P}{F} \frac{K_s K_m}{J}$$

$$W_t = \frac{126000 \times 100}{850 \times 3.518} = 4,214 \text{ lbs}$$

OVERLOAD FACTOR $K_o = 1.0$

DYNAMIC FACTOR $K_v = 8.5$ 780 ft/min

DIMETRAL PITCH $P = P_n \cos\psi$

$= 6 \cos 18°40'18''$

$= 5.68$

FACE WIDTH $F = 4.5$

SIZE FACTOR $K_s = 1.0$

LOAD DISTRIBUTION $K_m = 1.3$

FORM FACTOR $J_g = 0.55$

$J_p = 0.46$

TEMP. FACTOR $K_T = 1.0$

RELIABILITY FACTOR $K_R = 1.5$ HIGH RELIABILITY

LIFE FACTOR $K_L = 1.0$

RECONSTRUCTED CALCULATIONS

20 JAN 72

FIRST STAGE 66/1 RATIO

For Gear $S_{tg} = \frac{4214 \times 1.0}{0.85} \times \frac{5.68}{4.5} \times \frac{1.0 \times 1.3}{0.55}$

$= \underline{14,782}$ psi *

For Pinion $S_{tp} = 14,782 \times \frac{0.55}{0.46} = \underline{17,676}$ psi *

For Pinion $H_B = 300$ $S_{at} = 35,000$ psi ALLOWABLE TABLES

DESIGN STRESS $S_{ad} = \frac{S_{at} K_L}{K_T K_R} = \frac{35,000 \times 1.0}{1.5 \times 1}$

$= \underline{\underline{23,333}}$ psi. *

PINION O.K.

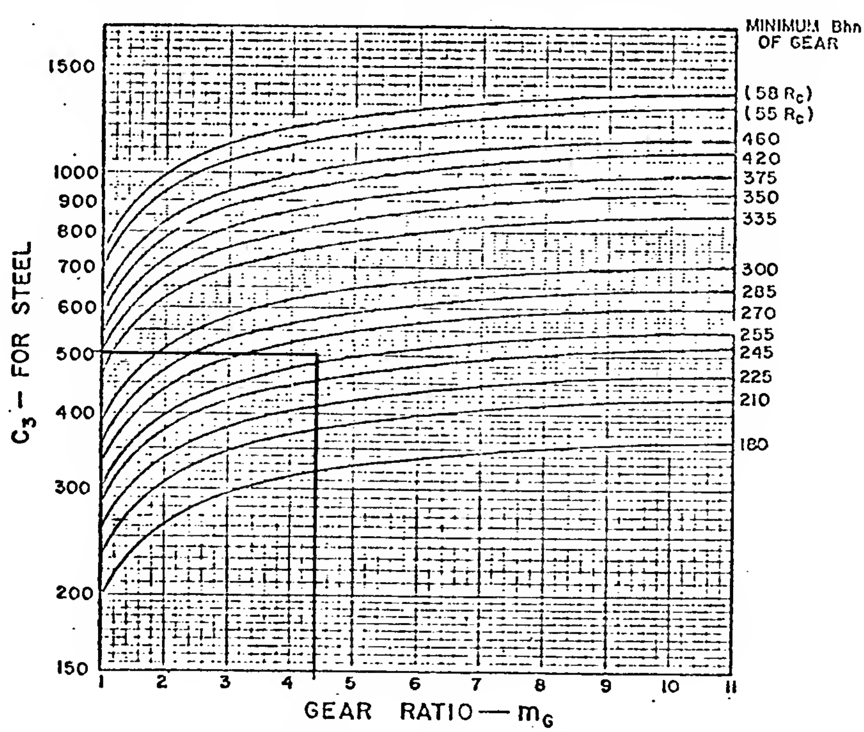
GEAR BASED ON WEAR O.K.

RECONSTRUCTED CALCULATIONS

VALUES ARE TO BE TAKEN FROM THE CURVE BELOW FOR THE MINIMUM HARDNESS SPECIFIED FOR THE GEAR. SUGGESTED GEAR AND PINION HARDNESS COMBINATIONS ARE TABULATED BELOW FOR CONVENIENCE. PINION AND GEAR MAY BE THE SAME HARDNESS WHEN THE GEAR RATIO IS 1.5 OR LESS.

AGMA 211.02A

	MINIMUM BRINELL HARDNESS												
GEAR	180	210	225	245	255	270	285	300	335	350	375	55 R_c	58 R_c
PINION	210	245	265	285	295	310	325	340	375	390	415	55 R_c	58 R_c



C_3 – VALUES FOR EXTERNAL GEARS

TABLE NO. 3

GEAR GEOMETRY

SHAFT NO.	1			
	FOR TOTAL RATIO 66 To /		FOR TOTAL RATIO 22 To 1	
DESCRIPTION	PINION	GEAR	PINION	GEAR
No. of Teeth	20	88	44	65
Pitch Dia.	3.5185	15.4814	7.6697	11.3302
Outside Dia.	4.0185	15,6481	8.0030	11.6636
Theoretical P.D.	3.6851	15.3148	7.6697	11.3302
Addendum	0.2500	0.0833	0.1666	0.1666
Dedendum	0.1489	0.3156	0.2323	0.2323
Hand of Helix	R.H.	L.H.	R.H.	L.H.
CTR Distance	9.5000		9.5000	
Face Width	4,500		4.500	
Ratio	4.400		1.477	
Helix Angle	18° 40' 18"		17° 01' 56"	
Sin of H.A.	0.3201453		0.2929089	
Cos of H.A.	0.9473684		0.9561403	
Normal P.A.	20°		20°	
Normal D.P.	6		6	
Backlash	0.005 - 0.010		0.005 - 0.010	
Ratio of add	3 to 1		1 to 1	

TABLE NO. 4

GEAR GEOMETRY

SHAFT NO.	II		III	
DESCRIPTION	PINION	GEAR	PINION	GEAR
No. of Teeth	26	94	20	83
Pitch Dia.	5.4166	19.5833	6.9900	29.0100
Outside Dia.	6.0166	19.7833	7.9900	29.3434
Theoretical P.D.	5.6166	19.3833	7.3234	28.676
Addendum	0.300	0.100	0.500	0.1667
Dedendum	0.1787	0.3787	0.250	0.583
Hand of Helix	L.H.	R.H.	R.H.	L.H.
CTR Distance	12.500		18.000	
Face Width	5.5		10.5	
Ratio	3.615		4.15	
Helix Angle	16° 15' 37"		17° 30' 09"	
Sin H.A.	0.280000		0.3007479	
Cos H.A.	0.960000			
Normal P.A.	20°		20°	
Normal D.P.	5		3	
Backlash	0.006 - 0.012		0.008 - 0.016	
Ratio of Add	3 to 1		3 to 1	

There are several gear steels available. Each has its own unique advantages and its own processing requirements. Three types of steel were considered for the gears of the coiler reducer:- alloy steel, carbonizing steel and nitriding steel.

Alloy steels are suitable for gears requiring high wear resistance, and high load carrying capacity. They are generally through-hardened. Through-hardening is a rather simple and economical process. If teeth are cut after heat treatment, the hardness is limited by machinability. The practical limit of hardness for free cutting leaded steel is about 350 BHN.

Carbonizing is a process where low carbon steel is subjected to a carbonaceous environment at temperature between 1650° and 1750° F, causing the carbon to diffuse into the steel. This produces a gear which has core properties different from the surface. The carbonized case can be harder than the core. The case after heat treatment ranges from 62 to 64 Rockwell C. The high surface hardness and favorable residual compressive stresses provide a high resistance to wear, pitting and fatigue. The core should have enough flexural strength to resist tooth loading and avoid case failure. Carbonized gears may be considerably smaller than through-hardened cut gears for equal load rating. Unfortunately, teeth tend to distort during the carbonizing and tooth grinding is usually required. Tooth grinding is a very specialized process, requiring special equipment, which increases the cost of gears approximately 25%.

Nitriding steels are used for gears where improved wear resistance, low notch sensitivity and satisfactory elevated temperature operation are requirements. This involves heating of the gears in an atmosphere of partially dissociated ammonia at temperatures of 950° to 1050° F. Quenching is not necessary.

Nitrided gears have exceptionally high case hardness, about 65 Rockwell C. Care must be taken to ensure that the core has sufficient strength to support the thin nitrided layer. Because of a relatively thin nitrided case (.015" to .020") gear surfaces are susceptible to crushing under overloads.

Two additional surfaces hardening processes were considered for the coiler reducer. These were induction hardening and flame hardening.

Induction hardening is particularly suitable for high volume production. High initial cost of heating coils coupled with low quantity ruled this process out.

Flame hardening was also ruled out because it is not generally satisfactory for precision gearing. It is utilized largely for large bull gears. The flame hardening process produces thermal distortion of gear teeth, tends to unwind the helix angle and produces residual thermal stresses at the roots of the teeth.

Because only two units were required, the use of existing manufacturing facilities and the most economic combination of materials and heat treatment dictated the choice of through-hardened alloy steel for the gear train.

Materials chosen consisted of SAE 4140 chrome-moly steel through-hardened to 270 - 300 BHN for the gear and SAE 43L40 chrome-moly-nickel, through-hardened to 310 - 340 BHN for the pinion. This alloy contains about 0.15 to 0.30% lead, which improves machinability without appreciably affecting the mechanical properties of the steel. There are many advantages to the use of through-hardened steel; economy and availability are obvious factors but of paramount importance is the high accuracy obtainable in finished gears, since no heat-treatment is required after cutting.

The relatively small outside diameter of pinions favoured pinions integral with shafts. Integral shaft and pinion sets were produced from forged through-hardened bars. These were rough machined to remove the decarburzing zone. Failure to do so could impair the wear resistance of

the tooth tip. After rough machining the pinion assemblies were stress relieved at a temperature of 850 to 900° F. to minimize the possibility of distortion. Excellent surface finish was obtained by the use of the lead bearing steel.

High speed and first and secondary train pinions were also machined from solid forgings for the same reasons. It has been our experience that where low volume production is involved, forgings are more economical for outside diameters up to 20 inches.

The low volume, delivery requirements, and the existence of excellent steel fabrication facilities resulted in the low speed gear being of welded steel construction. Cast steel gear blanks were not used because:

1) If the volume is low and the delivery requirements are stringent, the cost of pattern equipment and the time lost while the pattern is being constructed can be excessive.

2) Foundry errors such as blow holes, porous sections and cracks are inherent in any casting procedure. Although defects can be detected by various non-destructive methods such as x-ray checking or ultrasonic inspection, the process is time-consuming and costly. Repair of castings by welding is undesirable because of its influence on the metallurgy of the casting. Furthermore, many foundry errors such as slag inclusion can occur deep in the body of the casting, where remedial action is not practical.

   Although most reputable foundries will re-supply defective castings at no charge, the time lost is not compatible with low volume and quick delivery. If the casting flaw remains undetected until the cast blank has been largely machined, this will result in a considerable increase in manufacturing cost due to the need to re-machine a casting.

The low speed gear was fabricated from an alloy steel ring utilizing SAE 41L40 steel with webs and hub in type A36 mild steel. After welding the assembly was stress relieved at 850 - 900° F.

The shafts are subjected to stresses due to bending and torsion. A compressive stress exists, due to the thrust load generated by helical gears; this is uaually relatively insignificant and is normally disregarded. The rigidity because of deflection requirements of the shafts is so high that elastic stability is rarely critical.

AGMA CODE 260.01 permits use of the ASME formula for shaft design, but listed allowable strength is considerably lower than that suggested by the ASME.

Since the ASME formula involves very arbitrary factors and is not related to endurance strength in a direct manner, I prefer to use the ASME formula for preliminary shaft design only. If the stresses determined by this manner come close to AGMA allowable strength values, I would recalculate the shaft utilizing the Solderberg equation with due consideration for details such as fillet radius, keyways and other sources of stress concentration.

The final result of 3,925 psi maximum stress is considerably lower than the AGMA allowable shear stress of 10,800 psi. Consequently, the shaft satisfies design criteria for combined bending and torsional loading. Since it is appreciably lower than the AGMA allowable, there is no need to recheck the shaft as long as proper precautions are taken in shaft detailing to ensure that no undesirable stress concentrations exist.

The pinion deflection is composed of two parts, due to bending and torsion. Both act in the same tangential plane; therefore, the combined deflection is obtained by algebraic addition of these two deflections. AGMA specification No. 211.02 entitled Standard Surface Durability of Helical Gear Teeth, limits the combined bending and torsional deflection of the pinion to not over .001" over the entire face.

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INPUT SHAFT

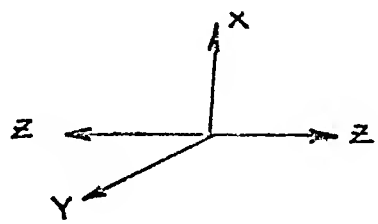
RECONSTRUCTED CALCULATIONS

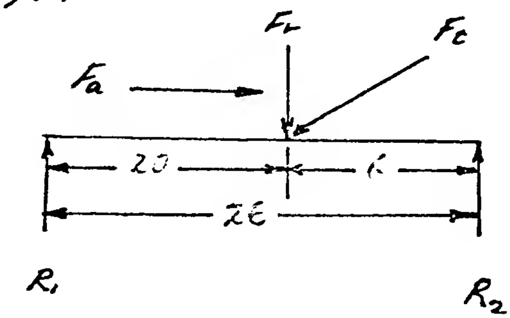
$HP = 100$; $N = 850$ R.P.M ; $d = 3.158$ in.

$\phi = 20°$ PRESS. ANGLE ; HELIX ANGLE $\psi = 18° 40'$

TORQUE $T = \dfrac{63025\ HP}{N} = \dfrac{63025 \times 100}{850}$

$T = 7414$ lbs/in





TANGENTIAL FORCE $F_t = \dfrac{T}{r} = \dfrac{7414 \times 2}{3.518}$

$F_t = 4214$ lbs

RADIAL LOAD

$F_r = \dfrac{F_t \tan \phi}{\cos \psi} = \dfrac{4214 \times 0.364}{0.947}$

$F_r = 1540$ lbs.

AXIAL LOAD

$F_a = F_t \tan \psi = 4214 \times .321$

$F_a = 1350$ lbs.

IN "Y - Z" PLANE

$R_1 = \dfrac{4214 \times 6}{26}$ $\qquad$ $R_2 = \dfrac{4214 \times 20}{26}$

$= 974\#$ $\qquad$ $= 3240\#$

BENDING MOMENT

$M_1 = 974 \times 20 = 19480$ lbs/in DUE TO F_t

PLANE "X-Z"

$R_1 = \frac{1540 \times 6}{26}$ $\qquad$ $R_2 = \frac{1540 \times 20}{26}$

$= 355^{\#}$ $\qquad$ $= 1185^{\#}$

BENDING MOMENT

$M_2 = 355 \times 20$ $\qquad$ $M_3 = \frac{1350 \times 3.518}{2}$

$= 7100$ lbs in DUE TO F_r $\qquad$ $M_3 = 2370$ lbs in DUE TO F_a

ASME FORMULA FOR COMBINED SHEAR STRESSES IN SHAFTS SUBJECT TO BENDING AND TORSION

$$S_s = \frac{16}{\pi d^3} \sqrt{(C_m M)^2 + (C_t T)^2}$$

S_s = ALLOWABLE SHEAR STRESS (DESIGN STRESS)
M = BENDING MOMENT
T = TORSIONAL MOMENT
C_m = BENDING MOMENT FACTOR (ROTATING STEADY LOAD 1.5)
C_t = TORSIONAL MOMENT FACTOR (STEADY LOAD 1.0)

$$M = \sqrt{M_1^2 + (M_2 + M_3)^2}$$

$$M = \sqrt{(19480)^2 + (7100 + 2370)^2} = \underline{21700} \text{ lbs/in}$$

$T = 7414$ lbs in

$S_s = 10,800$ psi $\qquad$ FROM. AGMA 260.01 FOR 300 BHN STEEL

SHAFT TO BE INTEGRAL WITH HIGH SPEED PINION MIN.
$d = 3.518$ P.D.

RECONSTRUCTED CALCULATIONS

RECONSTRUCTED CALCULATIONS

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INPUT SHAFT

$$S_s > \frac{16}{3.518^3 \pi} \sqrt{(21700 \times 1.5)^2 + (7414)^2}$$

$$> \underline{\underline{3920}} \text{ psi}$$

MUCH LESS THAN AGMA ALLOWABLE 10,800 psi !!!

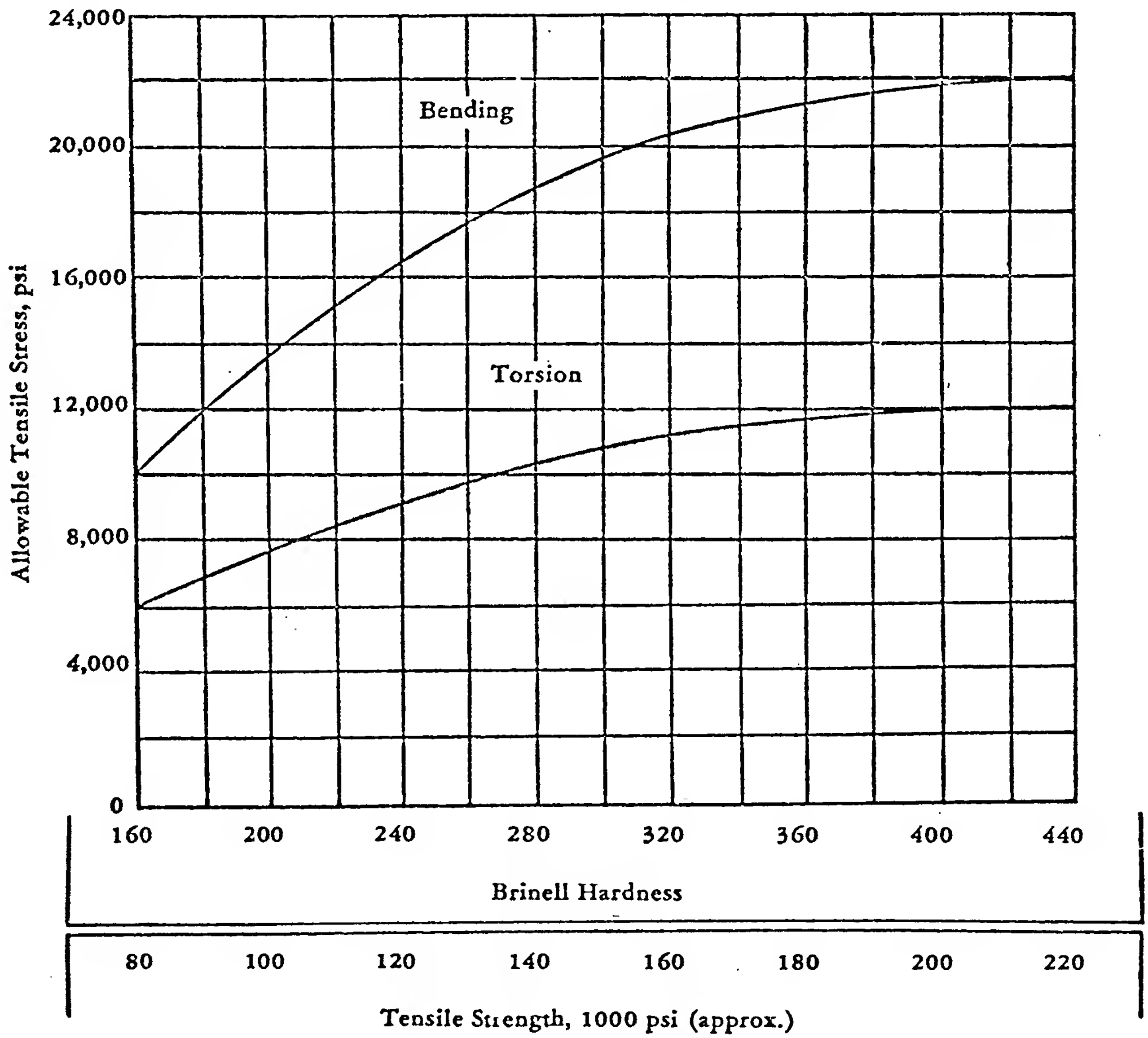

Figure 1

SOURCE: AMERICAN GEAR MANUFACTURERS ASSOCIATION
NO. 260.01

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PINION DEFLECTION

RECONSTRUCTED CALCUATIONS

BENDING DEFLECTION OF GEAR TOOTH

$$y_b = \frac{2}{E\pi} W K^4 \; n - \frac{7}{12}$$

SPECIFIC LOAD $W = 935$ lbs/in $\left(\frac{4214}{4.5}\right)$
ELASTIC MODULUS $E = 30 \times 10^6$
FACE WIDTH-DIAMETER RATIO $F/d = K = 4.5/3.518 = 1.28$
BEARING SPAN - FACE WIDTH RATIO L/F $n = 26/4.5 = 5.76$

$$y_b = \frac{2 \times 935 \times 1.28^4}{30 \times 10^6 \pi}\left(5.2 - \frac{7}{12}\right)$$

$$= .000247 \text{ in.}$$

TORSIONAL DEFLECTION

$$y_t = \frac{4}{G\pi} W K^2$$

TORSIONAL MODULUS $G = 11.5 \times 10^6$

$$y_t = \frac{4 \times 935 \times 1.28^2}{11.5 \times 10^6 \times \pi}$$

$$= 0.000170$$

TOTAL DEFLECTION

$$y = 0.000247 + 0.000170 = 0.000417 \text{ in.}$$

AGMA 211.02 ALLOWS 0.001 in ; DESIGN O.K. !!

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1 of 1 TOOTH CORRECTION.

RECONSTRUCTED CALCULATIONS

SIGG'S FORMULA FOR CORRECTION AT FIRST POINT OF CONTACT.

$\Delta u = 2 + (2.8 \times W \times 10^{-3})$ LOWER LIMIT TOLERANCE

$\Delta o = 5 + (2.8 \times W \times 10^{-3})$ UPPER LIMIT TOLERANCE

CORRECTION IN TEN THOUSANDTHS OF AN INCH.

$W = 935$ lbs/in

$\Delta u = 2 + 2.8 \times 935 \times 10^{-3}$

$= 4.62$ or 0.000462 inches.

$\Delta o = 5 + 2.8 \times 935 \times 10^{-3}$

$= 7.62$ or 0.000762 inches

STANDARD HOB WILL RELIEVE 0.001 INCHES OF THE TIP AND ROOT OF THE GEAR TOOTH, NO FURTHER PROFILE MODIFICATION REQUIRED !!

Three types of bearings were considered for the coiler reducer. There were taper roller, spherical roller and cylindrical roller bearings.

After a few preliminary checks, heavy duty cylindrical roller bearings and deep groove ball bearings were chosen. This choice was dictated by economics and a desire to maintain centre distances at a minimum.

Cylindrical roller bearings offer maximum capacity for pure radial loads and deep groove ball bearings proved adequate to resist thrust loads. A unique economical bearing assembly was designed to take advantage of bearing characteristics (Exhibit 10).

Radial bearings were selected using Hyatt catalogue data.

The thrust bearings, 6314 ball bearings, were selected according to SKF specifications. The thrust bearing had a 32,700 hour B10 life.

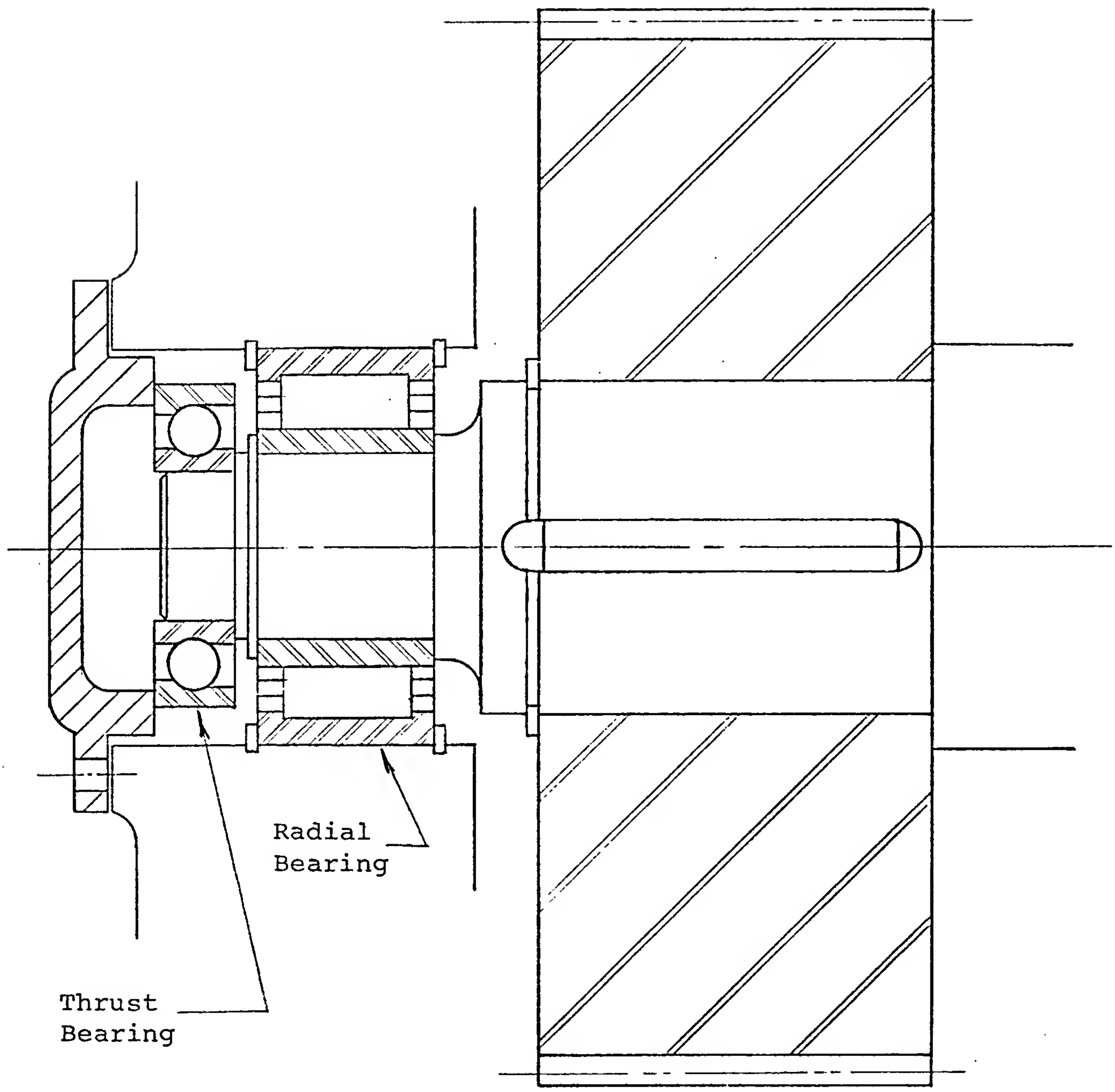

Exhibit 10

Bearing and Bearing Housing Detail

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1 of 1

BEARING SELECTION

RECONSTRUCTED CALCULATIONS

BEARING LOADS

PLANE "Y-Z" = 3240 lbs
PLANE "X-Z" = 1185 lbs
THRUST COUPLE = $2370/26$ = 91 lbs.

TOTAL BEARING LOAD $P = \sqrt{3240^2 + (1185+91)^2}$

$P = 3500$ lbs

REQ'D INPUT SHAFT $d = 2.750$ in min.

CHOOSE HYATT A5314-TS ROLLER BEARING
$d = 2.7559$ $D = 5.9055$

ASSUME ADJOINING SHAFT 2 HAS HYATT A5316-TS BEARING
$D = 6.229$
CENTER DISTANCE 9.5"

CLEARANCE BETWEEN BEARING $= 9.5 - \left(\dfrac{6.299 + 5.9055}{2}\right)$

$= \underline{\underline{3.4}}$" O.K.

CHECK BEARING LIFE

SPEED FACTOR S.F. = 0.563 (850 rpm)
CATALOGUE RATING C = 13,200 lbs (3000 hrs B10 @ 500 rpm)

$LF = \dfrac{13200 \times 0.563}{3500}$

$= \underline{2.12}$ LIFE FACTOR

FROM LF TABLES 2.12 → LIFE = $\underline{\underline{36,500}}$ B10 HRS.

AGMA SPECIFY 5000 HRS MINIMUM; O.K.!

A speed reducer's housing is subjected to internal and external loads. Internal loads result from transmitted (tangential) gear forces, separating forces and thrust loads. External loads are imparted by transmitted torque and overhung loads. Overhung loads exist when the output shaft or input shaft incorporates external gearing, chain and belt drives. Loads are transmitted to the housing through bearings, and external loads are balanced by forces at the housing mounting points.

A complete analysis of a gear housing is very complicated because it involves the analysis of multiple indeterminate structures, and the determination of movements and deflections in plates and shells. If analytical means are to be employed a computer is almost essential because of the work involved. Even then, the results must be used with considerable practical discretion. Factors of safety based on ultimate stresses must vary over a wide range, from the minimum of six to a high of twelve. Deflections must be limited to about .00005" per inch of bearing span.

The housing for the coiler reducer was designed on the basis of practical experience in the selection of housing thicknesses, hubs, and stiffeners for reducers of similar physical dimensions and load carrying capacity.

It is regrettable that competition dictates that design by experience must often replace design by analysis. Engineering cost can represent a large proportion of the total cost of the speed reduction unit. Consequently, low volume and the need for economy does not always permit the luxury of design by analytical or model analysis.

Housings for speed reduction units can be of castings, welded steel, or combined castings and welded steel design. Castings can be steel, ductile iron, or cast grey iron. Cast steel is normally 70 - 40 and ductile iron 60 - 40 - 03. Fabricated steel housings are normally made from steel plate to specification ASTM A-36 or the equivalent. The coiler reducer required a steel housing. Consequently, it remained to decide whether this steel

housing be a casting or fabrication. If the pattern is available, I have found that a cast steel housing is about 25% cheaper than the equivalent fabricated housing. However, a pattern was not available for the coiler reducer and it was necessary to analyse the cost difference between a casting and fabrication, as follows:

Cast Steel Housing:

Pattern charges:

Housing base	$1,876.00
Housing cover	$1,790.00
H.S. bearing retainers	$ 78.00
I.S. bearing retainers	$ 95.00
L.S. bearing retainers	$ 105.00
Total	$3,944.00

Cost of material:

Housing base approx.	4200 lbs.	@ $.47	=	$1,970.00
Housing cover approx.	3450 lbs.	@ .47	=	$1,620.00
(2) H.S. bearing retainers	18 lbs.	@ .75	=	$ 13.50
(2) I.S. " "	29 lbs.	@ .72	=	$ 21.00
(2) L.S. " "	68 lbs.	@ .64	=	$ 43.50
		Total		$3,668.00

For two units, cost per housing:

Pattern:	3944/2 =	$1,972.00
Material:	=	$3,668.00
	Total Cost	$5,640.00

Steel Fabricated Housing

Material A-36 steel approx. 7200 lbs.

Material cost plus handling charges	\$1,120.00
Labour including overhead	\$3,120.00
Stress relief	\$ 150.00
Sand blast	\$ 78.00
Total cost per housing	\$4,468.00

Difference between cast and fabricated housing:

Cast steel	\$5,640.00
Steel fabricated	\$4,468.00
	\$1,172.00

Total saving based on manufacturing cost:

2 x \$1,172.00 = \$2,344.00

The difference between the engineering cost of designing and detailing a cast or welded steel housing is minimal. The writer's experience indicates that the cost of engineering is roughly 2% greater where a fabrication is involved. This arises largely from the need to detail more pieces, and provide welding specifications. In light of the foregoing, the \$2,344.00 saving in the manufacturing cost ruled in favour of a housing of fabricated steel.

The thermal horsepower rating is the horsepower which a reducer can transmit continuously without overheating. Heat is produced due to gear efficiency, bearing friction, oil churning and windage.

Normally, speed reducers are operated safely with a temperature rise of 70° F, to a maximum operating temperature of 200° F.

The heat generated must be dissipated by the area of the housing exposed to the ambient air.

Many speed reducers have sufficient housing area to satisfactorily dissipate heat and retain thermal ratings within the range of mechanical ratings. If not, it is possible to increase the thermal ratings for smaller housings by utilizing auxiliary cooling. The most commonly used system is that of an integral fan mounted on the high speed shaft of the speed reducer. Other methods involve circulation of lubricant through a heat exchanger where the oil temperature is reduced by cooling water or air flow.

It is generally accepted that loss of efficiency per gear set in a helical train is in the order of 2% per reduction. The coiler reducer had three reductions, resulting in an overall loss of 6%.

Heat generated by gearbox -

$$\text{H.P. loss} = 100 \times .06 = 6 \text{ H.P.}$$

$$\text{OR } 42.41 \times 6 = 254.6 \text{ BTU/MIN}$$

The exposed area of the housing is 67.5 square feet. Based on test results and empirical formulae 50 square feet of exposed area will dissipate .0017 H.P. / square foot / $^{\circ}$F rise. Therefore, for a rise of 70° F the housing will dissipate $.0017 \times 67.5 \times 70 = 8.03$ H.P.

The thermal capacity of the reducer is considerably higher than required. Auxilliary cooling will not be required.

In enclosed gear drives, lubrication serves three basic functions. Lubricants separate tooth surfaces and prevent metal to metal contact, thereby reducing friction and wear. They also reduce heat generation of gears in mesh and carry heat away from gears and bearings. The main factors influencing lubricant selection are loads, speeds, operating temperature and reduction ratio.

Input power is of prime concern; the heavier the tooth load, the more viscous the oil must be to maintain a hydrodynamic oil film. Shock loads require a heavier oil than that required for uniform loads and special additives are often used to prevent breakdown of the lubricating film. Higher speed helps formation and maintainance of a hydrostatic oil film.

Ambient temperatures during start-up and operating conditions must be considered. The oil selected must not be so viscous that it fails to flow at low ambient temperatures. As temperature increases, viscosity decreases. The lubricant must be suitable for the anticipated temperature rise.

Overall reduction ratio influences lubricant selection. The speed of the first reduction set is higher than that of the second or third set. Since a thicker oil film is created at high speeds, the second or third reduction requires a heavier oil than that required by the first reduction.

Normally tooth loading is higher in the second and third reduction, therefore double and triple reduction units require a higher viscosity oil than a single reduction unit transmitting the same power.

A #4 EP (extreme pressure) oil was selected for the coiler reducer in consideration of the heavy shock loads that must be transmitted, range of input speeds, and the need for three stage reduction. This oil has a viscosity range of 700 to 1000 SUV at 100^{o} F. The fact that this lubricant had operated satisfactorily on similar applications in the past made it a logical choice.

Selection of the lubricant system was again dictated by economics. Splash lubrication is less costly, and was preferred as long as it represented good engineering practice. Since thermal considerations did not require oil to be circulated into a separate cooling reservoir, forced lubrication would not be a necessity unless dictated by the pitch line velocity. I have found that if pitch line velocities do not exceed 4000 fpm splash lubrication is satisfactory. The input ranges from 850 to 2000 rpm.

$$V = \frac{\pi\, dn}{12} \qquad \text{where } V = \text{velocity FPM}$$

$$n = \text{speed in RPM}$$

$$d = \text{pitch diameter}$$

$$V = \frac{\pi \times 7.669 \times 2000}{12}$$

$$= 4020 \text{ F.P.M. (maximum)}$$

$$V2 = \frac{\pi \times 7.669 \times 850}{12}$$

$$= 1695 \text{ F.P.M. (minimum)}$$

Since the anticipated maximum velocity was only slightly over 4000 F.P.M. it was decided that splash lubrication was a satisfactory and economical solution.

A study of lubricant flow through the various components of the speed reduction unit was made to determine where light gauge oil pans, oil grooves, flow directors and wipers should be located to reduce churning of lubricant to a minimum. These ancillary items served two purposes. They ensure an adequate supply of oil to all components over the range of operating speed and avoided increase in lubricant temperature due to agitation of the oil.

Lip type seals were selected for the coiler reducer because it is the most efficient in preventing leakage of free flowing lubricants. Labyrinth seals to operate in conjunction with the lip seals were not necessary because there was little danger of foreign abrasive particles being present outside the shaft area. However, the oil seals selected were of the spring loaded type with provision for grease lubrication. This type of seal was specified to ensure long life by virtue of lip to shaft interface lubrication. Besides reducing friction and heat generation at the seal, this type prevents the entry of dust and other contaminents by flushing the seal at regreasing intervals.

In the preparation of manufacturing drawings great attention was paid to the detailing of the shaft in the seal area. Surface finish is critical and the method of producing the finish is of vital concern. All seal contact surfaces were plunge ground rather than traverse ground to ensure that grinding marks were a series of parallel marks rather than spirals. To avoid possibility of seal failure due to sharp grinding marks, the surface was polished to 16 rms after plunge grinding. This effectively prevented seal lip abrasion and avoided pumping of oil along spiral grooves.

Another item that required attention was oil baffling. Helical gears have a tendancy to pump oil along lines parallel to the shaft centre line. This pumping can be a source of leakage if special attention is not given to the design of baffles that will absorb this pumping effect, and prevent it from reaching the seals. In the case of the coiler reducer, pumping baffles were mounted on the shafts in order to deflect oil back into the housing.

Oil bleeder grooves were also machined into the housing between bearings and seals. This ensured that excess oil between bearings and seals would be returned to the oil reservoir of the speed reducer.

The coiler reducer specification contained no specific noise limitations, there are many applications where it is essential to maintain noise levels at specified limits. Under these conditions the designer must pay strict attention to design considerations that reduce noise level.

Upon completion, the reduction units were shop tested under 50% rated capacity at maximum input rpm. The noise level recorded was 87 decibels measured at a distance of six feet from the speed reduction unit. This noise level was considered satisfactory for the application.

The resultant gear box (Exhibit 11 to 15) proved to meet the specifications and is now operating satisfactorily at the mill.

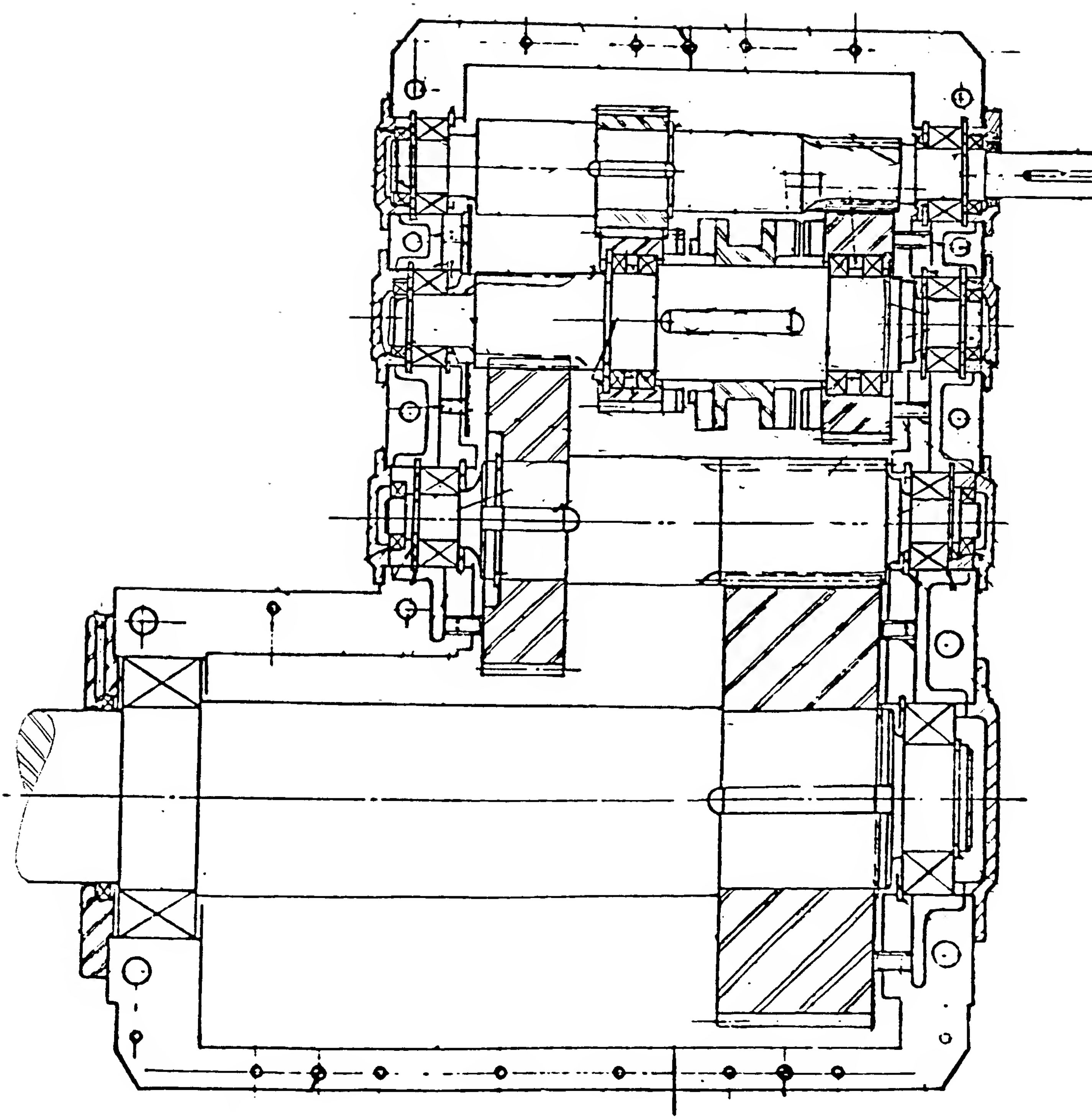

Exhibit 11

Gearbox Assembly Drawing

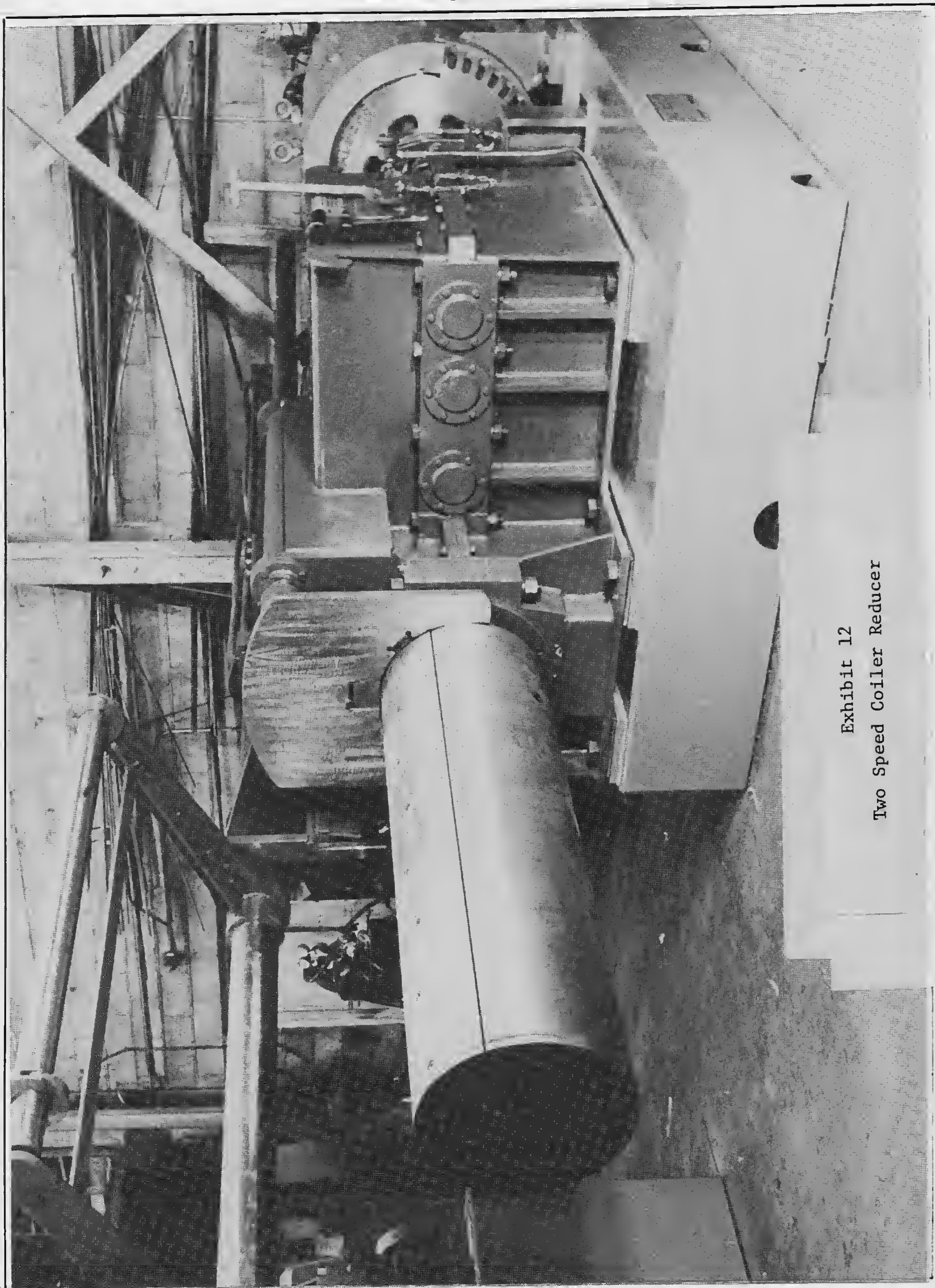

Exhibit 12

Two Speed Coiler Reducer

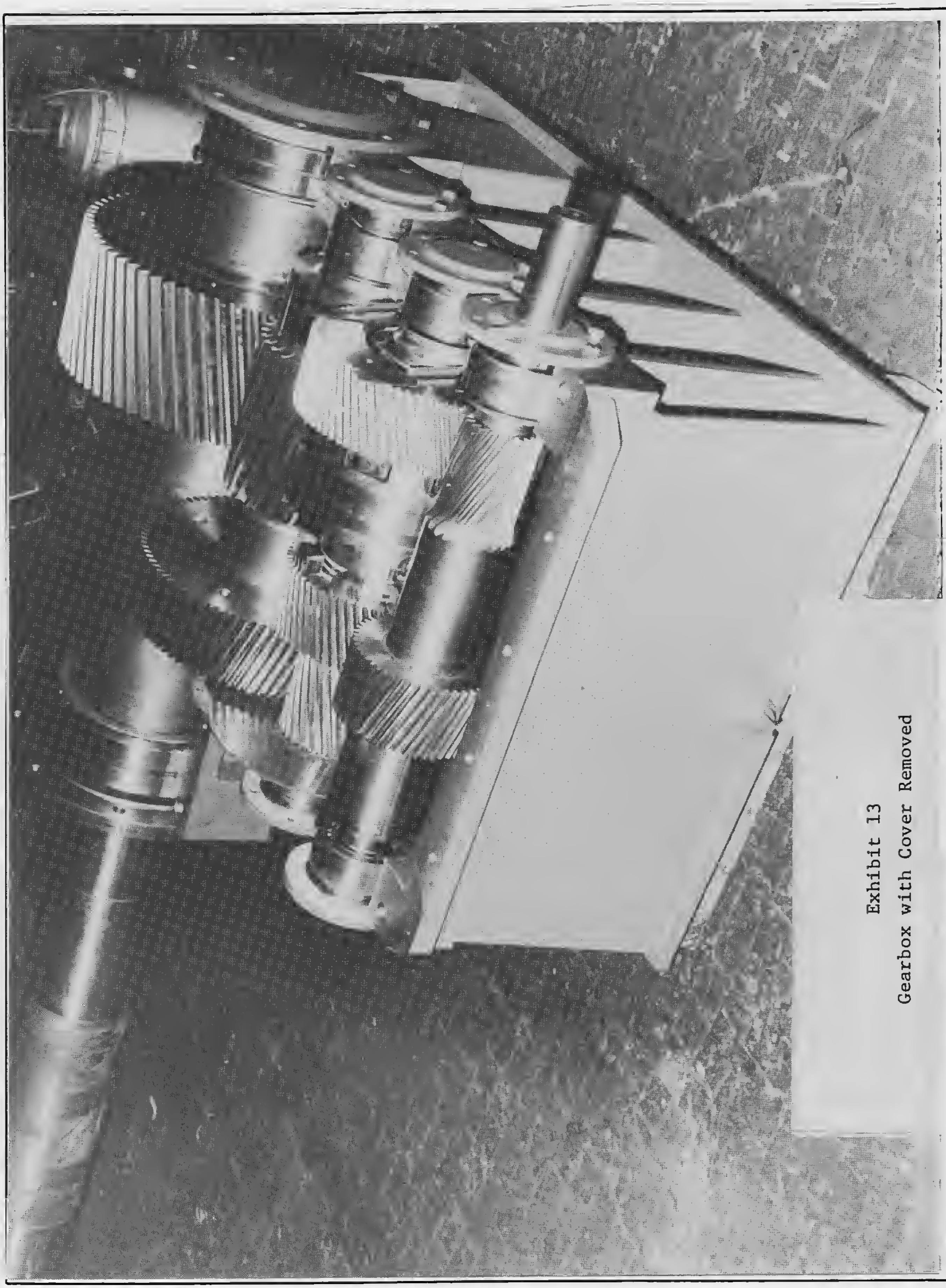

Exhibit 13
Gearbox with Cover Removed

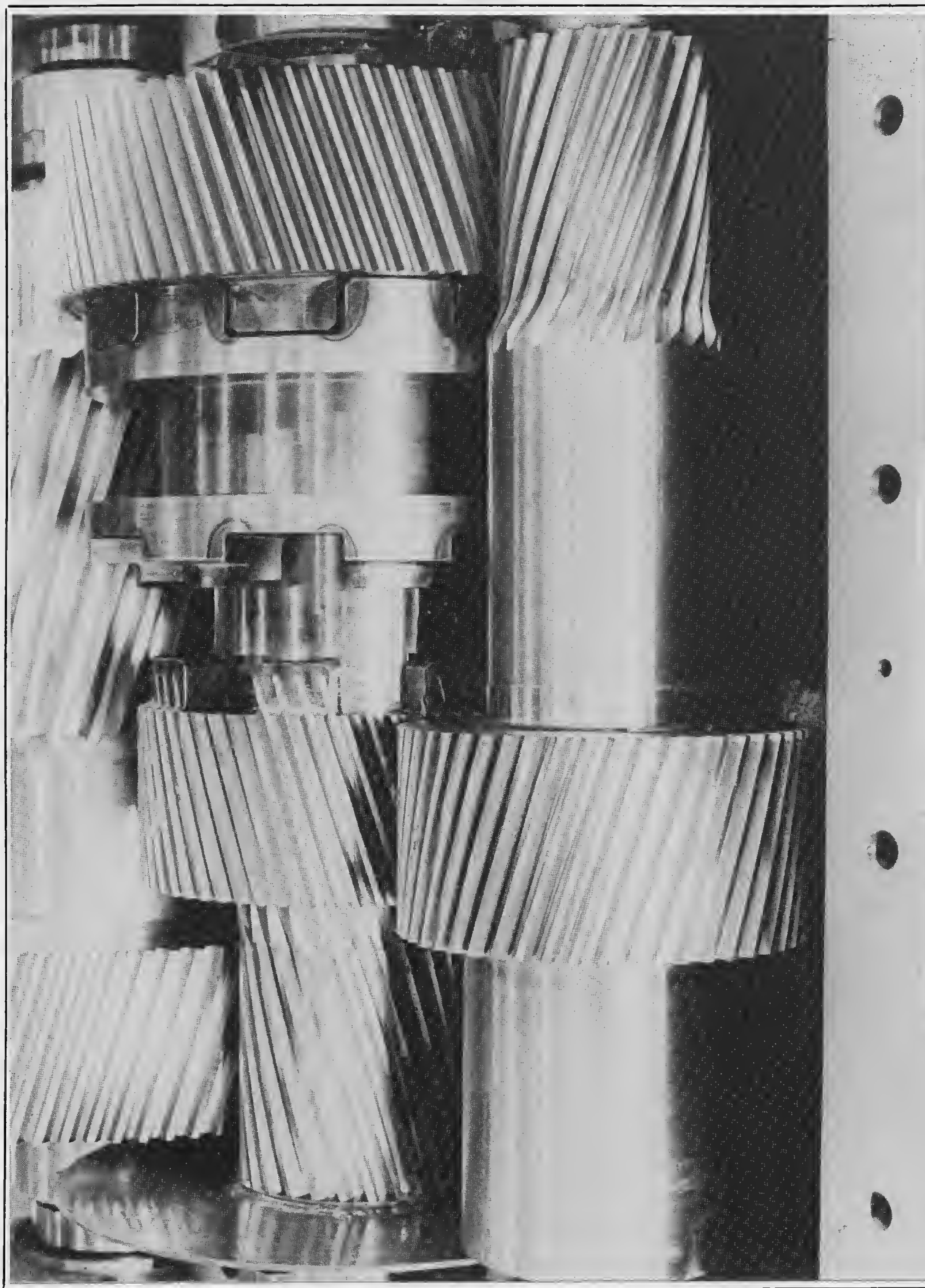

Exhibit 14

Detail of Clutch Arrangement

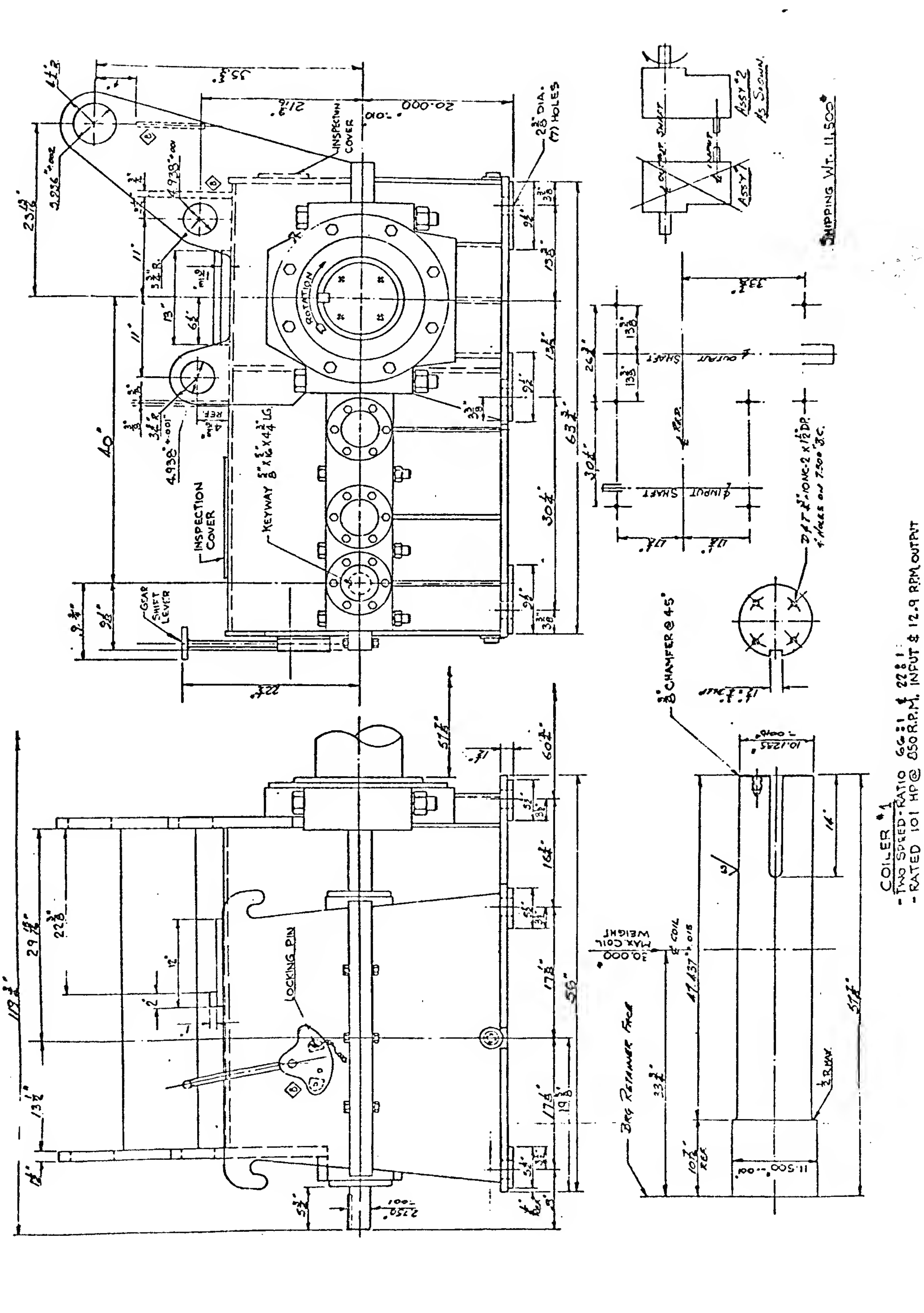

Exhibit 15

Outline of Two Speed Coiler Reducer

INSTRUCTOR'S NOTE

Mechanical Systems Design Project

The attached case Part A outlines the requirements for a two-speed uni-directional coiler gear box.

Your assignment is to design this gear box. You are to produce a layout drawing of sufficient detail that it could be given to a draftsman for detailing. A design report is to be produced justifying the choice of the principal components (gears, shafts, bearings, seals, etc.). This report should contain original calculations, NOT dressed up copies. Where necessary, freehand sketches of shafts, bearing arrangements, etc., should be in the design report.

The marking criteria for this project is given in the attached sheet.

A list of references that should be useful in making the necessary design decisions is attached.

Project to be delivered by:

Note:

This case was written as a source for a senior mechanical design project. Part A was issued with these instructions. After students had completed and submitted their designs, Part B was issued for discussion purposes.

FACULTY OF ENGINEERING

ECL 255N

Department of Mechanical and Aeronautical Engineering

Student ______________

Mechanical Systems Design

Because all of the projects submitted are N O T marked by the same person, the following marking criteria are to be used in assigning marks for the projects. In doing the evaluation consideration should be given to the time available in making the assessment, an incomplete good design may be worth more than a completed sloppy design.

The evaluation for each aspect of the project is on a 5 point scale.

	Unsatisfactory	Poor	Satisfactory	Good	Excellent	Marking Basis	Assigned Mark
1. **Overall Design** Will the unit perform the function satisfactorily as designed or with minor modifications ?						20	___
2. **Design Analysis** Has analysis been carried out on essential design parameters ? Have they been made correctly ?						30	___
3. **Secondary Design Feature** Have the secondary design features been properly detailed (Housing, Caps, etc.) ? Have assumptions been documented ?						20	___
4. **Documentation** Is the documentation clear and complete ? Does it reflect the design ?						15	___
5. **Assembly** Can the unit be easily assembled and overhauled ?						10	___
6. **Production** Can the components be readily manufactured ?						10	___

Total ______________

Marker ______________

Comment on five (5) things that were done well.

Comment on five (5) things that could be done better.

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